MECHANICAL LABORATORY
METHODS

KEITH

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MECHANICAL
LABORATORY METHODS

The Testing of Instruments and Machines in
Mechanical Engineering Laboratory Practice

BY

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PREFACE TO THE FIFTH EDITION

In presenting this new edition an effort has been made to preserve the basic ideas which Professor Smallwood maintained in the earlier editions. It is believed to be a sound principle that the mechanical engineering laboratory work should give the student the opportunity to apply the theory which he has studied in the classroom. This should be done without any "spoon-feeding" but, rather, should require considerable thinking on the part of the student and a minimum of assistance from the instructor.

No new tests have been added but the existing ones have been rewritten and improved where necessary. Lengthy derivation of formulas of thermodynamics, mechanics, etc., easily obtainable in the standard works, continue to be omitted from the text. A few new items have been added in Part I on Instruments.

The abridged tables of the properties of steam and ammonia have been deleted for two reasons: such tables are more of an aggravation than a help, and every student should be required to own and know how to use the complete tables and the accompanying Mollier Charts.

The A.S.M.E. Code on Definitions and Values has been revised and is available from that Society at a small cost. The abridged form of this code, included in the fourth edition, has been dropped.

While intended primarily for use in connection with undergraduate laboratory classes, it is hoped that it may be of value to the practicing engineer as well.

As in the previous editions, the effort has been made not to overburden the reader with descriptions of apparatus, operation and details of instruments or machines since these are better understood by examination of the thing itself. Particularly, descriptions of specific instruments have been kept in terms general enough to cover all of a given class or group. Fundamental principles are covered thoroughly so that the reader may apply them to a specific case.

<div align="right">F. W. K.</div>

February 1947

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MECHANICAL LABORATORY METHODS

INTRODUCTION

THE PRINCIPLES OF EXPERIMENTAL MEASUREMENTS

The science of experimental engineering rests primarily upon the art of measurement. Conclusions relating to general laws or specific operating conditions, formed from test results, stand or fall according to the accuracy with which the measurements have been made.

Absolute Accuracy. There is no such thing as absolute accuracy of measurement except by accident or chance. For instance, the length of a bar, measured with a scale graduated in hundredths of an inch, may appear to be 3 in. By chance, it may be that the bar is 3 in. long exactly, neither more nor less by even an infinitesimal fraction of an inch. This, however, is very improbable. An ordinary micrometer, capable of measuring to one-thousandth of an inch, may show it to be 3.005 in. long. An instrument of still nicer capability might yield another decimal place, and so on. It is evident, then, that absolute accuracy presupposes an instrument with graduations corresponding to infinitely small fractions of the unit of measurement.

Assuming that this condition could be sensibly attained, it would still be necessary to prove the correctness of the instrument within the value of its own graduations, and this could only be done by comparison with absolute standards of length; that is, material objects known to have definite linear dimensions under stated conditions of atmospheric pressure, temperature and method of support. But this implies measurement by an absolutely accurate instrument. Thus, to create such an instrument, it is necessary to have the very article desired.

Types of Errors. Not considering gross blunders, due to carelessness or ignorance, there are five types of errors which may affect the accuracy of measurements and thus prohibit absolute accuracy, as follows:

(a) Accidental errors.
(b) Personal errors.

(c) Instrumental errors.

(d) Errors in sampling.

(e) Errors due to improper location of instruments.

Accidental Errors. Accidental errors are all those which cannot be eliminated by instrumental or other corrections. They are due entirely to chance and are not systematic except that they follow the law of probability. Such errors are incurred in the calipering of a cylinder; when the calipers are, inadvertently, held at a slant instead of on a true diameter, when the calipers are held with varying degrees of pressure or when temperature changes momentarily affect the instrument.

Accidental errors are likely to be either positive or negative from which it follows that, in a series of observations, the probability is that the errors of excess will equal those of deficiency, *provided a sufficient number of readings are taken.* In general, the variation of the mean from the true value will be reduced as the number of observations is increased, but needless increase is to be avoided. According to the probability theory, it can be shown that the probable error of the arithmetic mean of a series of observations is equal to the probable error of a single observation divided by the square root of the number of observations in the series. In other words, the arithmetic mean of four determinations of a quantity will have a probable error only one-half as great as that of a single observation. From this it can be seen that, to reduce the probable error to $\frac{1}{100}$, ten thousand observations of the quantity will have to be made. Obviously such a procedure is wasteful of time and effort.

Personal Errors. This type of error arises from the fact that a particular observer, by reason of his personal characteristics and idiosyncrasies, may habitually read too high or too low.

Errors * due to the personal equation of the observer are of many sorts and in a sense are not subject to correction. Some observers are noticeably quicker than others and will therefore make more exact observations when dealing with quantities in a state of flux. Some observers consistently read a given instrument high and others consistently low. Some observers can estimate readings between scale divisions more exactly than others. Some observers have so persistent a memory for past observations that they unconsciously allow this memory to influence subsequent observations, while others seem to be absolutely unaffected in this respect. No method can be prescribed for guarding against errors arising from this source. The best that can be done is to distribute the available observers to the best advantage and to check the performance of each before the test or periodically during the test by having several observers make the same determinations independently.

* A.S.M.E. Code on General Instructions, Power Test Codes.

Since practically every experiment involves visual observation at some stage, such sources of personal error as defective eyesight improperly corrected or color blindness on the one hand, or defective lighting or unduly close instrument graduations on the other, may entirely vitiate the results.

Instrumental Errors. The precision, accuracy and sensitivity of all instruments should be examined carefully before using them for experimental purposes. *Precision* refers to the fineness of the graduations, *accuracy* to their correctness, and *sensitivity* to the response of the instrument to small changes in the quantity being measured.

The value of the smallest graduation of an instrument is called its *least count*. Calibration of the instrument does not affect its least count; correction factors determined by calibration merely correct the indications of the instrument and so bring them to the degree of precision of which the instrument is capable.

The accuracy of an instrument is established only by a comparison with some "standard unit," itself a copy or a copy of a copy of the "primary standard" kept at the National Bureau of Standards in Washington, D. C. Details of this sort of testing are given in Part One, but it is well to emphasize at this point that the necessary closeness of the approximation, of any standard used, to the primary standard depends entirely upon the least count of the instrument to be compared with it. Generally the standard is sufficiently accurate if it is correct within one-fifth of the least count of the instrument being checked. An accuracy of one-tenth the least count is commonly encountered. The least count of the standard need not be less than one-half that of the instrument tested.

The word *standard*, as used here, must be accepted in a relative sense. For some engineering work, secondary standards may be improvised which are just as useful as the most accurate standard possible. Good judgment should dictate when a given standard is adequate for the work in hand.

Again, although the instrument may have the properties mentioned previously, it may be used under such circumstances that it cannot respond to changes, in the quantity being measured, as rapidly as the changes occur. This is called the *sensitivity of the instrument with respect to time.* On the other hand, the instrument may be so constructed or in such a condition that, having come to a state of rest, it will require a change greater than its least count to cause it to register a new indication. This is called the *sensitivity of the instrument with respect to displacement or motive impulse.*

It should be evident that the precision of an instrument is not attainable if it is not sufficiently sensitive in both of these respects.

Errors in Sampling. Errors of this type are likely to occur when it is difficult or impossible to obtain samples that are representative of the bulk. Sampling of fuels and exhaust gases in combustion experiments or tests are instances where such errors may be encountered.

Errors Due to Location. Errors of this type are due to inability or inconvenience in placing or installing instruments so that they will respond to average conditions. A thermometer, for instance, might be placed so that it was out of the main stream of the medium being measured. Obviously the indications of the thermometer would not measure the real temperature of the medium but only of that portion with which it came in contact. Errors of this type can usually be eliminated or minimized by a careful study of the possible locations for instruments in a given experimental set-up.

Per Cent of Error. Variations from the true value or mean value are referred to in percentage. For example, a given observation is said to be correct within plus or minus 1 per cent. It is convenient to interpret this in terms of *significant figures* since it makes the degree of accuracy more obvious. A significant figure may be defined as any digit of a number which serves to indicate the magnitude of the number and not to locate the position of the decimal point.

When a result is represented by three significant figures the error occurring through the omission of a fourth figure is less than one-half of 1 per cent. Thus, the quantities 1010, 101.0, etc., may be correctly expressed 1015, 101.5, etc., but in each case the error is less than one-half of 1 per cent if the fourth digit is omitted.

Use of Significant Figures. Consider the five quantities:

(a) 256
(b) 0.00256
(c) 25.06
(d) 25.60
(e) 25,600

The quantity (a) has three significant figures which we interpret as meaning that the value is closer to 256 than it is to 255 or 257 or that it lies somewhere between 255.5 and 256.5, but that we are not sure of the value of the fourth digit and, therefore, have omitted it. Quantity (b) is an identical case, the two zeros to the right of the decimal point merely serving to locate it.

The quantity (c) has four significant figures as also does quantity (d). In the case of (d), however, there is some doubt as to the exact meaning of the final zero unless it is expressly stated that values are being reported to four significant figures. The final zero, which may have been recorded through ignorance or carelessness, implies that greater precision has been attained than was actually the case.

The quantity (e) is ambiguous because the two final zeros can be interpreted in two ways: (1) that the zeros are not significant and aid only in locating the decimal point, or (2) that they are significant and the value lies somewhere between 25,599.5 and 25600.5. If the first interpretation is intended it is better to write 2.56×10^4 or 256×10^2.

Rounding Off. In most engineering measurements we cannot secure results with less than 1 per cent error, so that three significant figures are ample for their presentation. Generally, the numerical value having the minimum number of significant figures determines the maximum number of significant figures entering into calculation of results and also indicates the number to be retained in the result. In a statement of results of a test it is useless and often misleading to give significant figures which indicate a greater degree of accuracy than that actually warranted by the precision of the instruments used or the degree of refinement in the method of testing.

When dropping significant figures to bring a numerical value into accord with the accuracy of the numerical value having the minimum number, it is usual to retain one more significant figure than the minimum, until the final result is obtained. This tends to prevent possible cumulative error.

When rounding off by dropping significant figures these rules should be followed:

1. If the first digit to be dropped is less than five, the last digit kept is left as it is.

2. If the first digit dropped is greater than five or is five followed by digits greater than zero, then the last digit kept is increased by one.

3. If the digit dropped is five followed by zeros only in the succeeding places, the number kept is rounded to its nearest *even* value.

For example, if it is desired to round off the number 256.45 to three figures the result is 256. Had the number been 256.55 the result would have been 257. The special case under rule 3 is illustrated by the number 256.50 which would be rounded to 256 since that is the nearest *even* value.

Intrinsic Evidence of Accuracy.   Assuming instrumental correctness equal to the least count, there are two ways of securing greater accuracy of measured results: first, by using an instrument of greater precision and sensitivity; and second, by making numerous repetitions of the measurement.   This latter is one of the greatest aids to accuracy in experimental engineering.   It is only by repeated trials that accurate conclusions can be framed because, not only are there variations which result from erroneous measurement, but the quantities measured are generally variable ones, themselves subject to change.

Under such circumstances, a single determination proves nothing and indicates little.   On the other hand, a series of determinations not only reduces the probable error of the result but provides evidence of its accuracy.   Consider, for example, a series of four measurements made upon a single constant quantity by observer A.   His observations are recorded as 103, 98, 101, 98 of which the arithmetic average is 100.

Assuming 100 to be the correct result (it is the best obtainable consistent with the observations, although, of course, not correct), then the error of each observation is the difference between it and the mean, or $+3, -2, +1, -2$, the plus and minus signs indicating whether the errors make the individual determinations greater or less than the mean.   Now, if another observer, B, measures the same quantity with the results 101, 100, 102, 101, averaging 101, the errors of his determinations will be 0, $-1, +1, 0$.   It is not difficult to see that B's measurement and result of 101 are probably more accurate than A's because his errors, figured from the mean, are less than A's.

A Warning to Beginners.   There is a tendency, on the part of beginning students in laboratory work, to overestimate their ability to interpolate between the graduations of an instrument, which often leads to a false sense of accuracy that is not justified.   For example, take the common type of steam pressure gage which is graduated in 5 psi. intervals with rather heavy lines and a pointer whose tip is approximately $\frac{1}{16}$ in. wide.   It is not at all unusual to see a student record a reading of such a gage as, say, 105.6 psi.

Let us examine the case to see if his value is legitimate.   The distance between graduation lines is about $\frac{3}{8}$ in. while the lines themselves are about $\frac{1}{32}$ in. wide; they are purposely made this way so as to be easily visible at some distance.   Assuming these dimensions to be accurate, $\frac{1}{10}$ psi. would be represented by a distance of 0.375 divided by 50 or 0.0075 in. Note that the width of the pointer, about 0.06 in., is some eight times the

width of the division our student has purported to have estimated. Obviously he has tried to do the impossible.

If he had recorded the gage reading to the nearest whole pound, he would have been doing very well indeed, but the tenths are out of the question. Estimating to the nearest $\frac{1}{5}$ of a division is about the best that can be done in instruments where the width of the graduation lines
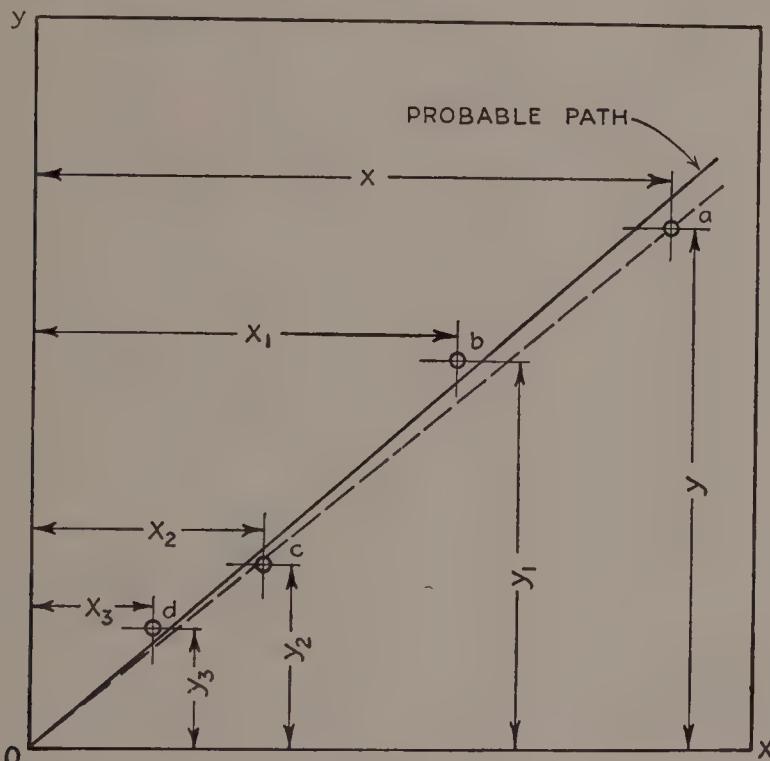
FIG. 1.

and the indicator or pointer are extremely small parts of the space between graduations. If these are quite appreciable fractions of the space, as was the case with the steam gage, the estimate is certain to be still less accurate.

Compound Quantities. Most engineering measurements are upon compound quantities, the components of which are variable. The following parallel illustrates the case and also the graphic method to be described later. Fig. 1 represents the plan of the floor of a room; and the solid diagonal line, the path of a ball that rolls across it. It is desired to locate as exactly as possible the path as it is traversed by the ball. The only exact data we have are that the path is a straight line and that it starts at O.

At a given instant one observer measures the distance of the ball from the wall OX and another from OY. These distances are x and y and are sufficient to locate one of the positions of the ball and therefore its path, provided they are accurately determined. Errors, of one kind or another, prevent this so that the path is falsely located on the line Oa. Obviously, if a number of points, instead of only one, could be located, a better result would ensue. The points b, c and d are the result of such determinations. It is found that these points do not lie on the same straight line, the distances from the true path being the errors of their determinations.

The question now arises as to how these discordant observations can be made to agree upon one result, which, although perhaps not the true one, is in best accord with the data. It is possible to locate a different line with each pair of observations, x and y locating one, x_1 and y_1 locating another, etc. One might conclude the use of the averages of the x's and y's would be the best, but it can be shown by the theory of errors arithmetical means do not yield the best results upon indirectly determined quantities. In our problem, the result is obtained by measuring two other quantities; the determination is therefore indirect. The most *probable* result is a line so located that the sum of the squares of the distances from it of the points a, b, c and d shall be a minimum. Such a line can be determined mathematically, but the process is sometimes rather arduous. For this reason such lines are more frequently drawn in by eye judgment. In this connection a warning note should be sounded; there is an almost unconscious tendency to give undue weight to end points of the plot even though it is realized that the intermediate points are probably just as reliable. Where particular difficulty is experienced in making observations to obtain the data for the end points, this tendency is very undesirable because, in such a case, the end points are not likely to be as reliable as are the intermediate points.

Variables, Independent and Dependent. A large part of engineering experimentation is similar to the simple illustration given in the previous section. Two variables, bound together by a more or less rigid law, are measured, and from the results the law is deduced. The law is not always represented by a straight line, but often by curves of one sort or another. *Whichever it is, it is the business of experimentation to discover.*

One of the variables can always be controlled; the other then follows it according to the law binding them. For example, when a spring is extended by a force, there are two variables—namely, the force and the extension—which, combined, give the law of the spring. In their measurement we may add predetermined increments of weight (standards) and

let the extension vary as it may, or we may add weight enough to cause predetermined increments of extension and let the weight increase as it may. The predetermined quantity is called the *independent variable;* the other, *dependent.*

In most experimentation there are other quantities which also might be considered as variables but it is best practice to limit the investigation to the two that are related as explained. For example, changes of temperature would conceivably affect the extension of the spring but, if we allowed the temperature to change as well as the force applied, it would be impossible to tell how much of the extension was due to the force and how much to the temperature change. Therefore, while investigating the relation between force and extension, the temperature must be kept constant. If the effect of temperature were desired, it would be necessary to keep the force constant and vary the temperature by suitable means.

It is not always easy to decide what the independent variable shall be. In some cases it is more convenient to control one quantity than the other which makes the choice more or less automatic. In more complicated experiments, arbitrary choice of the independent variable does not always lead to the best results and considerable judgment and experience are required.

Representation of Results by Graphs. It is almost always desirable to present experimental data as points on a graph and to join them, as nearly as possible, with a smooth curve. This is done by adopting a pair of rectangular axes, OX and OY, giving each a suitable scale according to the units measured. In Fig. 2, for instance, 1 in. along OX represents 10 lb. applied to a spring and 1 in. along OY represents ½ in. of spring extension. The observations 7.5 lb. and ⅜ in. are thus shown by the point indicated, and so on.

If the relation between the variables appears to be satisfied by some curve, other than a straight line, such a one is drawn. It is drawn as a smooth curve and almost always without any reverse curvature. Reverse curvature usually indicates more than two variables. The curve is usually drawn by eye as judgment dictates, but occasionally the more laborious method of least squares must be employed.

If any point, of a plotted series, lies markedly off the line or curve, it may be concluded logically that its determination contained some large error or possibly a mistake in plotting. If a check shows no error in plotting, then the point may be thrown out of consideration entirely. As

a corollary, the value of a set of measurements may be judged by the closeness of the plotted results to a smooth curve. It is evidence as to the accuracy with which the measurements have been made.

Conventions for Plotting. Engineering practice has developed certain conventional methods of graphical representation which should be fol-
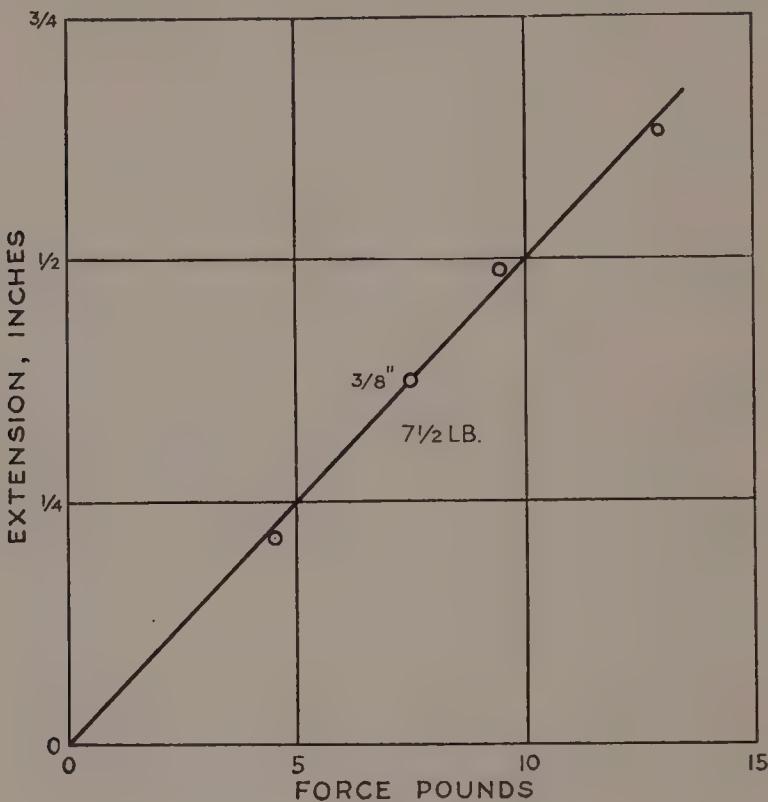
FIG. 2.

lowed as rigidly as possible in the interest of uniformity since it will aid greatly in making comparisons with the work of others. The more important rules for making graphs are enumerated as follows:

1. Graph paper is usually the imprint of an engraving with clear borders around the edges. The beginner is very prone to use the edges of the engraving as axes and to place the numbering of the scales in the border. This is not good practice because the border space is usually so limited as to crowd the numbering and labeling of the scales and, if the graph is bound into a folder, the lettering may disappear into the binding.

Draw in a pair of axes indented sufficiently from the edge of the engraving to allow for numbering and labeling of the scales.

2. The independent variable is always plotted along the abscissa, the dependent variable on the ordinate. The only exceptions to this rule are correction curves and stress-strain diagrams. Correction curves are plotted with the instrument reading as abscissa and corrections as ordinates. Stress-strain diagrams have deformations plotted as abscissas.

3. Choose convenient scale-factors for each of the scales, always using multiples of 2 or 5. Scales, whose subdivisions come out at 2.5 or 3.33, are very troublesome when it is desired to read off intermediate values from the curve to say nothing of the difficulty in accurate plotting of the original points.

4. The plotted points should be clearly marked by drawing a small circle about each point. These should be drawn with a compass for neatness and should be not more than $\frac{3}{32}$ in. in diameter. If points for more than one curve are to be placed on the sheet, or the data of different observers is to be indicated, a differentiation can be made by using small squares, triangles or other symbols in addition to the circles which are to be preferred.

5. The best smooth curve is drawn in through the points but the line of the curve should not be allowed to pass through the circles or other symbols surrounding the points. The purpose of this is to leave the points visible so they may be checked at any time.

6. Calibration curves and correction curves are drawn with small sections of straight line joining the points. This is done because the errors are mostly accidental and do not conform to a mathematical law. Obviously the points of calibration should not be too far apart but at fairly frequent intervals.

7. Always draw curves to as large a scale as possible and still have the curves well centered on the sheet. It is not always necessary to show the origin; to have the scales start at zero.

8. Do not place too many curves on one sheet, particularly if the curves cross one another, as this leads to confusion and a cluttered appearance.

9. Exaggeration of one scale, with respect to the other, sometimes accentuates the errors of the points to such a degree that it is difficult to determine the path of the curve. This is a precaution to be observed in connection with (7). On the other hand, a decrease in the scale, used on the ordinate, will often clarify the path of the curve. This will become more apparent as experience is gained.

General Rules for Testing. The following general rules should be observed when making measurements upon variable quantities:

1. All instruments and testing apparatus should be calibrated before the test and recalibrated or checked after the test. If there is any doubt that a given instrument will retain its calibration throughout the test, it should be checked at intervals during the test. In such cases the readings should be corrected on the basis of the average of the calibrations immediately preceding and following the observations.

2. Whenever possible, one or more preliminary trials should be made to determine the adequacy of the instruments and apparatus and to acquaint the personnel with their duties. The value of this practice is obvious.

3. The person in charge of the test should have a sufficient number of assistants so that he may be free to give special attention at any point where he may be needed. He should make sure that all instruments and testing apparatus are giving reliable readings and that the observations are being properly made and recorded. He should see to it that the chosen operating conditions are maintained.

4. Prior to beginning the test, or before continuing the test after important changes in the operating conditions, the apparatus should be run for a sufficient length of time to establish equilibrium in the new conditions.

5. Observations of fluctuating quantities should be made frequently and at equal time intervals to secure a better average.

6. The number or frequency of observations, which it is necessary to make of any quantity, depends upon the constancy of the quantity. The more variable the quantity the more frequently the observations should be made. If all the conditions of a test could be kept absolutely constant, only two sets of readings would be necessary. The two sets would give identical results; the second set would be made only to check the accuracy of the first. In some few cases this condition is approximated but these are the exception, not the rule. The exercise of considerable judgment is required to determine if such practice is compatible with the results to be attained. Ordinarily it is best to make frequent intermediate readings.

7. Tests involving time-rates should be extended over a sufficient period so that the errors of initial and final measurements will be reduced to 1 per cent, or less, of the total quantity.

8. In time-rate tests, intermediate readings should be taken to check or prove the constancy of the rate. Such readings also strengthen the determination against mistakes.

9. In making time-quantity measurements, it is always best to record the time of day rather than only the time interval. This insures a correct record of the time; otherwise it is very easy to note an interval of 3 min. when 2 or 4 min. have actually elapsed. A further advantage is that the readings of one observer may be readily correlated with those of another observer and possible irregularities in results may be accounted for by related happenings that can be associated only by a knowledge of the time of occurrence.

10. Every event connected with the test, however unimportant it appears at the time, should be recorded with the time of occurrence. Particular care should be taken to record any adjustment of the apparatus under test, whether made during a run, or between runs. There should also be a statement of the reasons for making such adjustments.

11. It is desirable to make rough calculations on the data as it is accumulated, during the progress of the test. Such calculations serve to indicate omissions, irregularities and mistakes, and they have the particular advantage of making these evident at the time they occur so that they may be rectified at once and often save a test which might otherwise be of no value.

12. At the completion of a test, all records should be assembled and reviewed to determine how they are to be used in working up the results. Any overlooked irregularities which appear should be eliminated or corrected. All such adjustments should be noted and explained in the final report.

Working Up the Data. After reviewing the records, the next step is to determine the method of calculation to be employed. Assuming that the data are expressed in a form compatible with their accuracy, it is possible to treat the individual repeated measurements in such a manner as to determine the most probable value and also to estimate the degree of accuracy and reproducibility which has been attained. The methods of doing this are:

(a) The Arithmetic Mean.
(b) The Geometric Mean.
(c) The Mode.
(d) The Median.
(e) The Weighted Mean.
(f) The Mean Square.

To illustrate the meaning of these various terms, let it be assumed that we have n quantities: $X_1, X_2, X_3, \ldots X_n$. The arithmetic mean of these quantities is the sum of all the X quantities divided by n. The

geometric mean is the nth root of the product of all the X quantities. The mode is simply an arbitrary selection of that value of X which appears the greatest number of times in a given set. The median is the middle value of a set of quantities arranged in order of ascending magnitude if the number of quantities is odd. If n is even, the median is taken as a value half-way between the two center values. The weighted mean is not used very much since it depends to a large extent upon the judgment of the person employing it. In this method certain values, believed to be more representative, are multiplied by constants such as 2, 3 or 4, etc. The sum of these products is then divided by the sum of the multipliers.

Under certain conditions it can be shown that the best probable result is obtained by use of the mean square. This is the nth root of the sum of the squares of the n quantities. This is a rather laborious process and, for most engineering work, the arithmetic mean is probably the best for general use.

The Report. Any report of the results of an engineering test should be complete and self-explanatory. It should always begin with a short statement of the objects of the test. This should be followed by a brief statement or résumé of the results which were obtained. In other words, these two statements should be, in effect, "We set out to do this; this is what we found."

Next should come a complete presentation of the facts of the test containing a description of the apparatus or equipment under test, the instruments and other apparatus used in making the measurements, their location and calibration. Sketches can be used to good advantage in connection with this part of the report. This portion should also include a discussion of the conditions of operation and also the names of the personnel conducting the test.

The next section of the report contains tabulations of the results of the test as calculated from the records and data taken at the time of the test. Included here are the graphical representations of the data in the form of plotted curves. Following these is the discussion of the results both as to accuracy and bearing on the object of the test. Following this can be included any leading conclusions which may be drawn from the results.

In the back of the report are placed the appendices which contain copies of all the original data taken at the time of the test, calibrations of instruments, descriptions of any special apparatus used, results of

any special or preliminary tests and an outline of the methods used in making the computations of the results.

The Power Test Codes Committee of the American Society of Mechanical Engineers has prepared several general and numerous special codes relating to testing work. These have been prepared in the interest of uniformity and should always be consulted where work of any magnitude is being undertaken. The Code on General Instructions should be read by everyone who is doing test work.

PART ONE

THE TESTING OF INSTRUMENTS

WEIGHTS AND FORCES

Weights and forces are measured by comparison with known weights with the aid of a system of levers such as in a beam balance, or by reference to the amount they deform some elastic object, as a spring, which has been previously calibrated against standard weights. When a leverage system is used so that a large force may be measured by comparison with a small known weight, it is necessary to know the ratio of the lever arms. In Fig. 3, for instance, the force F is measured by by the relation
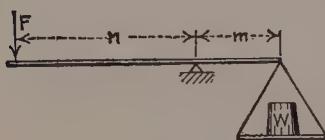
FIG. 3.

$$W = \frac{n}{m} F$$

in which n/m may be referred to as the "leverage ratio."

The calibration of any force-measuring apparatus rests primarily upon the force of gravity as a standard. "Standard weight" is a misnomer in that the *weight* of the body so called, being the force of gravity between it and the earth, actually varies with the locality of the body. Standard mass is a better term. A standard mass establishes a standard force when the acceleration of gravity at the given locality is known.

1. CALIBRATION OF PLATFORM SCALES

Principles. The platform scales, Fig. 4, is arranged so that a large weight, W, to be measured, may be balanced by a small weight on the beam, B. The small weight is either or both P on the poise or R, the rider, acting at a variable distance from the beam fulcrum. If P balances W, then by the principle of moments,

16

$$P \times \frac{a}{b} = F = \frac{W}{2} \times \frac{d}{c} + \frac{W}{2} \times \frac{e}{f} \times \frac{g}{c}$$
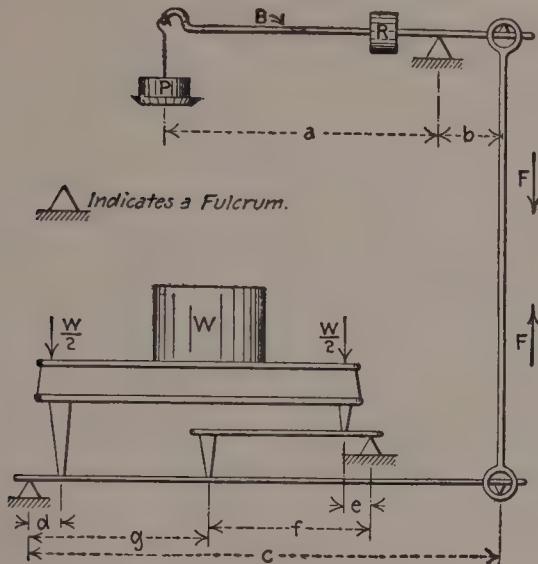
FIG. 4.—Platform Scales.

It is arranged so that $\dfrac{d}{c} = \dfrac{e}{f} \times \dfrac{g}{c}$,

$$P \times \frac{a}{b} = W \times \frac{d}{c}$$

and

$$\frac{W}{P} = \frac{ac}{bd} = \text{the leverage ratio.}$$

It will be noted that the zero position of the rider, R, is not directly over the fulcrum. When there is no load on the platform and the beam is balanced with R in its zero position, it should be obvious that R is balancing a portion of the mass of the platform and levers. The principles of mechanics will show that this force cancels out for any other position of R and, hence, the position of R along the beam is a true measure of the change of load on the platform. The proof is left to the reader.

(a) **The Sensitiveness of the Scales.** This is determined by placing a large weight on the platform, weighing it, and then finding the smallest additional weight that will cause a deflection of the beam which can be nicely balanced by the rider. This additional weight is also a measure of the precision of the scales, provided the beam graduations are small enough to take cognizance of it.

(b) **Beam Calibration.** Readings of the beam are taken corresponding to a number of standard weights. These should be sufficient to cover fairly the range of the beam. If standard weights of convenient size are not available, a number of small weights may be standardized for the purpose by using a calibrated scales with slightly greater precision than that of the scales to be tested.

If the instrument readings do not agree with the true weights, a calibration curve should be plotted, having the true weights as abscissas and the instrument readings as ordinates. This curve may be used to determine corrected values at any part of the beam.

(c) **The leverage ratio** may be found by measurement of the lever arms, but this method is difficult and subject to error. A better one consists of balancing a standard weight on the poise with a standard weight on the platform, then calculating the ratio of these weights. The nominal leverage ratio may be learned by examination of the poise weights. They are marked with their actual weights and the weights they are intended to balance, as, for instance, 1 lb.–100 lb. This gives a nominal leverage ratio of 100. To test the accuracy of this, place, for example, a standard ½-lb. weight on the poise and a standard 50-lb. weight on the platform. If they do not balance, add enough weight (shot is convenient) to either one or the other until a balance is secured. This additional weight may then be accurately measured and the true leverage ratio found.

(d) **Poise Weight Calibration.** The indications of the poise weights are accurate if the leverage ratio is true and if the actual weights of the poise weights are as marked. Therefore, they should be weighed by a calibrated scales of sufficient precision. It should be noted that any error in the poise weight is multiplied in the instrument reading by the leverage ratio. The standard scales should therefore weigh these weights to within an amount equal to the precision of the scales to be tested divided by its leverage ratio.

If the actual weight of a poise weight is not as marked, or if the leverage ratio is not true, then the weight balanced on the platform is

Actual weight of poise weight × actual leverage ratio

instead of the amount indicated by the marking. For instance, if the poise weight intended to measure 100 lb. actually weighs 0.99 lb. instead of 1 lb., and if the leverage ratio is 99.5 instead of 100, then the weight on the platform balanced by it is $0.99 \times 99.5 = 98.5$ instead of 100 lb. In this way the true weight balanced by each poise weight may be found.

(e) **Beam Calibration by Test of Rider.** In some types of platform scales there are no poise weights, a number of riders on different beams being used. Other types make use of a poise weight in conjunction with a rider, or riders, to give intermediate readings. In such cases, the following method may be used:

The weighing beam to be tested is represented by Fig. 5. With no load, the rider is in the position shown, its pointer being at the zero graduation, and its beam floating. With a load W on the table, the rider must be moved through a distance d to secure balance. If the scales are correct, the weight of the rider must be such that the distance
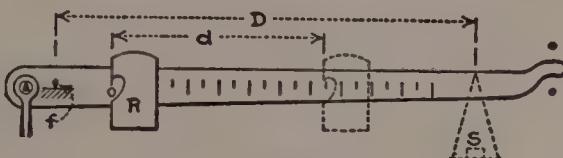
FIG. 5.

d will be that between the zero and the graduation marked W. Now, instead of using the rider, another weight could be applied at any convenient part of the beam as shown by S, of such amount as to secure a balance when the load W is on the table. Therefore, the moment of the weight S about the fulcrum f must equal the moment of the rider which is replaced. Letting R denote the weight of the rider,

$$S \times D = R \times d.$$

Let the ratio of the load on the platform to the balancing weight S be L. Then

$$S = \frac{W}{L}.$$

Substituting this in the first equation, and simplifying, we find the weight which the rider must have in order to suit the existing graduation of the beam:

$$R = \frac{D}{d} \times \frac{W}{L}.$$

To apply this relation to the calibration, a value of W is chosen, and the distance between the beam graduation marked W and the zero graduation is carefully measured. This determines W and d. A point is then chosen to represent D, and its distance measured from the fulcrum f. To find the ratio L, it is not necessary to apply the full weight W to

the platform, since the ratio $W:S$ is constant for all values of W, the ratio of levers being established by the selection of D. Therefore the following procedure is used. An appreciable, but not inconvenient, number of weights is placed on the platform, these having been previously measured with a standard scales. They may consist of anything available that may be readily moved. A balance pan, improvised from pasteboard and wire, is then attached to the beam at the distance D from the fulcrum, and to this pan are added shot or other small weights until a balance is secured. If desired, the weight of the pan may be previously balanced by the usual beam counterweight so that the experimenter need deal with the added weight only. The latter should then be accurately weighed. This result, divided into the weight applied to the table, gives the desired ratio L. The value thus obtained completes the data necessary to calculate the weight of the rider according to the equation previously deduced. The rider is then removed from the beam and weighed; if the actual checks the calculated weight, the instrument is proved at the graduation marked W. Any other graduation may then be checked by proportion, since the distances of the graduations from the zero mark must vary directly with the indicated loads.

Some types of scales have one or more riders fitted with thumb screws so that the riders may be clamped in any given position within the limits of movement. In this way a rider may be set to cancel out the tare weight of a container in which bulk substances are being weighed. The weight of the thumb screw is part of the weight of the rider and should the screw be lost the indications on the graduated beam will be inaccurate.

Most platform scales are designed so that eccentric loading has no effect. However it is well to investigate this since a dull knife-edge in one corner might cause eccentrically placed loads to be weighed inaccurately.

If the scales are to be moved a great deal or are apt to receive rough handling, it would be well to determine the effect of such treatment on the accuracy before proceeding with the actual calibration. If the scales are sensitive to eccentric loading or are easily thrown out of calibration by the rough handling of ordinary use, they should be repaired or discarded.

2. Evaluation of a Spring

Principles. In a large class of instruments, it is necessary to know the amount of force required to extend, compress, or twist a spring per unit of deformation. This quantity is called the "spring scale." According

to Hooke's law, it is a constant within the elastic limit of the material. In many cases the movable end of the spring is attached to some combination of links and gears designed to indicate, magnify, or translate the motion. When the applied force is increasing, the friction of such links or gears makes the instrument read low, since the indicator is moving up the scale and friction tends to hold it back. With a decreasing force, the instrument reads high for a similar reason. Suppose an external force F is applied to a spring instrument, which is correct except for the effect of friction, first by increasing the external force to the value F, and then by decreasing it to this value. Then, if S is the spring scale, E_1 and E_2, the extensions in the two cases, and X the friction,

(adding)

$$F = S \times E_1 + X$$
$$F = S \times E_2 - X$$

$$2F = S(E_1 + E_2)$$

$$F = \frac{S(E_1 + E_2)}{2}.$$

That is, the effect of friction may be eliminated by taking increasing and decreasing readings at each load applied, dividing the sum of the extensions thus found by two, and using the result as the extension caused by the external force.

(a) **Spring Scale by Graphic Method.** Take as an example the following measurements of force and extension.

Force, Pounds	Extensions		
	Increasing, Inches	Decreasing, Inches	Average, Inches
10	0.32	0.34	0.33
20	0.64	0:70	0.67
30	1.02	1.08	1.05
40	1.32	1.36	1.34
50	1.68	1.70	1.69

These are plotted as points on coordinate paper; then draw what appears to the eye as the best straight line to satisfy each of the three

sets of points.*   Fig. 6 shows the line for the increasing readings; the other two curves are similar.   When it is desired to apply a calibration only to increasing readings, the ascending line is used, and similarly with the descending.   The combined line gives the best results when the instrument is used for measurement of a fluctuating quantity.
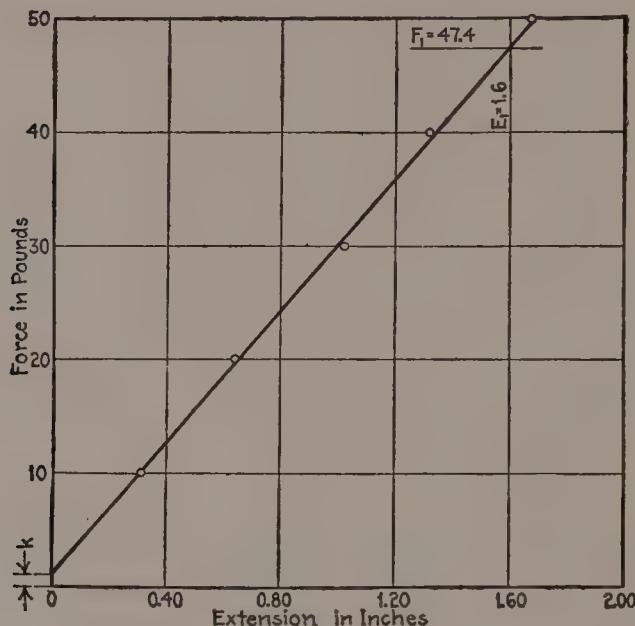
FIG. 6.

To figure one of the spring scales, as for instance the ascending, the line for which is shown by Fig. 6, a point is selected near its extremity, as F_1E_1, this point not necessarily representing a pair of observations. It should, however, lie *on* the line.   Then the spring scale equals

$$S \text{ (ascending)} = \frac{F_1 - k}{E_1}$$

or, using numerical values,

$$= \frac{47.4 - 1.0}{1.60} = 29.0 \text{ lb. per in.}$$

to be used when the readings of the instrument increase; similarly with the other two spring scales.

* Certain researches of the author indicate that the decreasing values of force and extension do not follow Hooke's law within appreciable amounts and that, theoretically, they must follow a curved line.   (*Physical Review*, October, 1911.)   To represent them by a straight line is therefore merely a convenient approximation.

Note that the intercept, k, is taken into account. If the curve does not pass through the origin it is because of friction or backlash of the indicating mechanism or a false zero reading of the extension.

(b) **Spring Scale by Method of Least Squares.** The theory of errors shows that the most probable straight line to fit such data is one so located that the sum of the squares of the distances of the plotted points from the line shall be a minimum. This can be calculated as follows. The equation of the line may be written

$$F = SE + k$$

in which k and S are unknown, and F and E are represented by a number of more or less discordant observations. This equation may be multiplied through by the coefficient of each unknown, thus

$$F = SE + k.$$

$$EF = SE^2 + Ek.$$

In each equation the corresponding values of E and F are substituted, thus forming two series of equations. Each series is summed and the resulting equations are known as the "normal equations" of k and S, respectively. These may be solved as simultaneous equations to obtain the most probable value of S.

Using the data of the ascending scale tabulated,

$F = ES + k$	$EF = E^2S + Ek$
$10 = 0.32S + k$	$3.2 = 0.1024S + 0.32k$
$20 = 0.64S + k$	$12.8 = 0.4096S + 0.64k$
$30 = 1.02S + k$	$30.6 = 1.0404S + 1.02k$
$40 = 1.32S + k$	$52.8 = 1.7424S + 1.32k$
$50 = 1.68S + k$	$84.0 = 2.8224S + 1.68k$
$150 = 4.98S + 5k$	$183.4 = 6.1172S + 4.98k$
Normal equation of k	Normal equation of S

From these, by eliminating k, it is found that $S = 29.38$ lb. per in.

The descending scale is found similarly and the combined scale is obtained by adding the corresponding normal equations from the ascending and descending values.

Note that the numerical values in the equations should be figured to four or five significant figures since, when the equations are solved, the

first two or three figures disappear by subtraction. Labor can be saved by using multiples of ten, or simple figures, for the independent variable F, and by using a pocket-book table of squares to obtain the values of E^2.

PRESSURE

The measurement of pressure is a special case of force measurement, wherein the area over which the force is distributed is taken into account. Pressure is a compound unit being the number of units of force per unit area.

In calibrating pressure-measuring devices, the standard force is that of gravity acting on some standard mass.
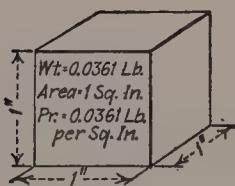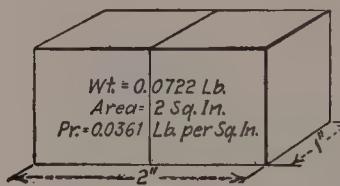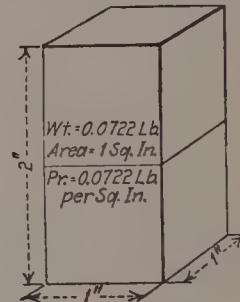
FIG. 7.   FIG. 8.   FIG. 9.

Pressure Produced by Water.

Pressure is usually expressed in pounds per square inch. Its measurement is always relative—that is, it is based upon some other pressure as a datum. A pressure gage reads so many units *above* atmospheric pressure, for instance, and a vacuum gage, so many *below*.

Another measure of pressure is the height of water or other liquid which by its weight balances the pressure to be measured. Fig. 7 represents a cubic inch of water. At 60° F., this weighs 0.0361 lb. Since the area this force acts upon is one square inch, the pressure produced is 0.0361 psi. Two cubic inches of water, arranged as in Fig. 8, would have twice the weight, but it would be imposed upon twice the area; hence the pressure would be the same. Arranged as in Fig. 9, however, the area would be one square inch, and therefore the pressure would be twice that shown in Fig. 7. It follows that the pressure produced by a given mass of water varies directly with its height and is independent of its cross-section. Hence, inches or feet of water may be regarded as units of pressure.

Fig. 10 shows the application of this principle in the "manometer." The pressure in the chamber C is said to be "X inches of water" and this equals $0.0361 \times X$ psi. The absolute pressure in C equals this quantity plus the pressure of the atmosphere represented by the arrow at A.

Various other liquids are used in manometers according to the range of pressure to be measured. Liquids heavier than water which are commonly used are tetrabromoethane (density 2.97) and mercury (density 13.6). Liquids less dense than water are various alcohols and oils. The tetrabromoethane, mercury and oils have the advantage that they do not mix with water and thus can be used to measure pressure in conduits carrying water.

The pressure, corresponding to a given difference of height in a manometer, can be changed to pounds per square inch by multiplying 0.0361 by the density of the liquid used in the tube. Thus mercury represents $0.0361 \times 13.6 = 0.491$ psi. per inch of height. If the density of a fluid is not known it may easily be determined by weighing a given volume of the fluid. This is best accomplished by use of a specific-gravity bottle obtainable from any laboratory supply house.

FIG. 10.—Manometer.

When water is used as the gaging fluid in a manometer, it may be rendered more visible by adding a minute amount of water-soluble *fluorescein* to the water. This produces a yellow-green color easily seen in the tube. The amount of fluorescein necessary to give a good color is so small that the density of the water is not noticeably changed.

When manometers are used to measure pressure of other liquids, it frequently happens that this liquid flows into the leg of the manometer connected to the vessel or conduit in which the pressure is to be measured. When this happens, a false indication is produced in the manometer. A correction, however, can be applied to obtain the true reading. The correction is calculated by the equation

$$\text{Correction} = h \times \frac{d_1}{d_g}$$

where h is the uncorrected manometer reading;

d_1 is the density of the liquid being measured;

d_g is the density of the gaging liquid.
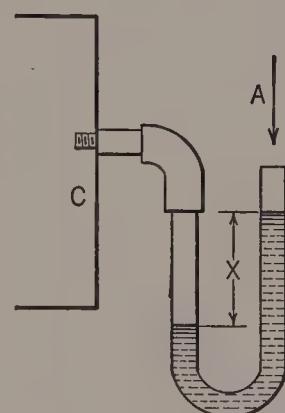
The correction is additive if the pressure is negative, and subtractive if positive.

Manometers of the U-tube type made of uniform bore tubing may be regarded as primary gages which require no calibration except that the density of the gaging fluid be known with considerable accuracy.

Another type of manometer is the cistern manometer shown in Fig. 11. In this type, the level of fluid in the cistern falls as the level rises in the manometer tube. If the diameter of the cistern is made quite large, compared with the diameter of the manometer tube, only the reading of the level in the tube need be taken. The reason for this can readily be seen from the following example:

Suppose that the cistern diameter be made 16 times the tube diameter, then the error introduced by the fall of the level in the cistern will be of the order of 0.4 per cent. With still larger cistern-tube diameter ratios the error is further reduced. The manometer can of course be calibrated against a standard or by computation in order to obtain true readings.
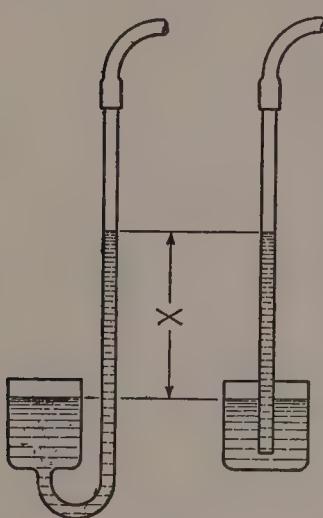
Where pressure differences must be measured with considerable accuracy, use may be made of the **sensitive differential manometer,** sometimes called a **micromanometer.** This instrument depends upon the use of two liquids of only slightly different specific gravities. As shown in Fig. 12, it consists of two cisterns connected by a U-tube. A liquid of specific weight w_1 is contained in both cisterns, whereas a liquid, of specific weight w_2 which is slightly greater than w_1, is placed in the U-tube.

When a change of level occurs in the U-tube, there will be a small accompanying change of level in the cisterns in the ratio of a/A, where a is the cross-section area of the U-tube and A is the cross-section area of each cistern. Thus, if H is the difference of level in the U-tube, the displacement of level in each cistern, h, will be:

$$h = \frac{aH}{2A}.$$

If d is taken as the distance from the mean level of the cistern liquid down to the lower leg of the liquid in the U-tube, we may write

Fig. 11.

$$P_B + w_1\left(d - \frac{aH}{2A}\right) + w_1H = P_C + w_1\left(d + \frac{aH}{2A}\right) + w_2H$$

from which,

$$P_B - P_C = (w_2 - w_1)H + w_1\frac{aH}{A}.$$
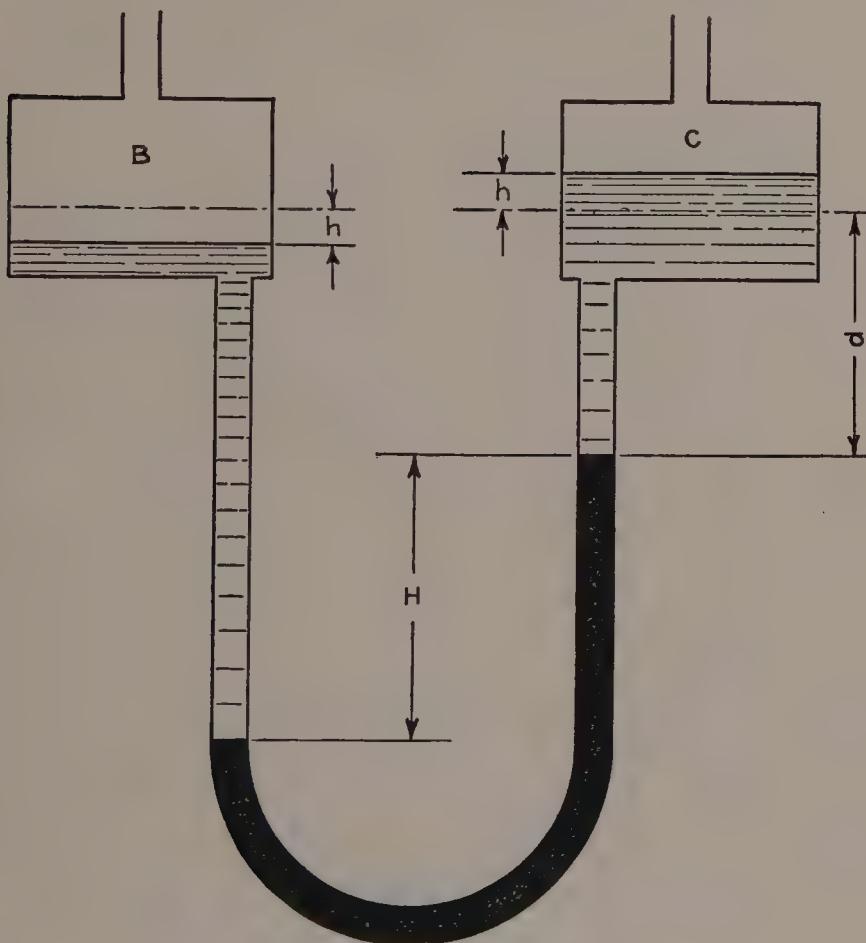
FIG. 12.—Micromanometer.

If the diameter of the cisterns can be made very large with respect to the diameter of the U-tube, the last term becomes negligible and may be omitted.

A special form of the cistern manometer is the draft gage used to measure the reduced pressure (less than atmosphere) in chimneys, which creates the draft. This is expressed in inches of water and, as it is usually only a few tenths, the ordinary U-tube will not do. The draft gage has one leg on a slant, so that the travel of the liquid is magnified. In use, the instrument should be set to its true designed level and the liquid

adjusted so that the level in the inclined tube stands at zero when there is no difference in pressure.

Because a draft gage depends upon the slant of the tube for its indications, it is not a primary gage. Some draft gages are made to be used with a gaging fluid of a definite specific gravity and yet to indicate directly in inches of water. For these reasons, draft gages should be calibrated against some sort of standard before being used.

3. Calibration Draft Gages

Calibration of a draft gage can be made by examination of its graduations and the weight of the liquid, or by comparison with a standard instrument. (See Fig. 13.)

Fig. 13.—Draft Gage.
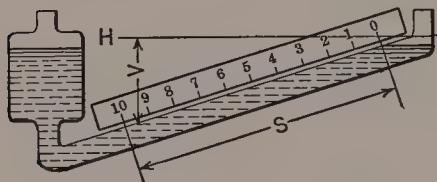
(a) **Calibration by Calculation.** Having set the gage to its designed level, a horizontal line *H–H* is drawn with a spirit-level through the zero of the scale. The vertical distance *V* from the last graduation on the scale to this line is then measured. The level of the liquid in the inclined tube will fall this distance for the whole scale, but the level in the enlarged tube will rise an amount equal to the length *S* of the scale in inches multiplied by the ratio of the squares of the diameters of the bores of the small to the large tube. If water is the liquid, the sum of the rise and fall thus calculated equals the inches of water pressure. The true pressure corresponding to any other graduation can now be found by proportion.

The calculation of the rise of level in the enlarged tube assumes that the bores are uniform.

Fig. 14.—Arrangement for Testing.

Commercial forms of this type of gage generally use oil for the liquid, having a definite specific gravity. When such a gage is calibrated, the total vertical difference of level is found as before. The corresponding height of water is then found by dividing this by the specific gravity of the particular liquid used. The specific gravity may be obtained by using a hydrometer or by weighing a known volume of the liquid.

(b) **Calibration against a Standard Gage.** Connect the gage to be tested with a piece of rubber tubing to the standard gage as in Fig. 14.

The pressure in this tube may readily be reduced to any desired amount by applying the lips to the branch B, the pinch-cock serving to confine the suction when obtained. Enough readings for a calibration curve should be taken.

For a standard gage, an impromptu instrument is easily made by using an inclined tube of generous length.

4. BAROMETERS

Mercury Barometers

In order to obtain absolute pressures it is necessary that the pressure of the atmosphere be known. Atmospheric pressure is measured by an instrument known as a barometer. Two general types are recognized—the mercury barometer and the aneroid barometer.

about 30 in

FIG. 15.

The mercury barometer is a special adaptation of the cistern manometer with the upper end of the tube sealed. (See Fig. 15.) Before being inserted in the cistern, the tube is completely filled with clean mercury. When the tube is placed upright, with its lower open end dipping into the mercury in the cistern, the mercury in the tube falls back slightly to a height of about 30 in., at sea level, and an almost perfect vacuum is formed in the upper closed end of the tube. The column of mercury in the tube is, of course, sustained by atmospheric pressure acting on the surface of the mercury in the cistern. Changes in atmospheric pressure cause the level in the tube to change. Since there is a nearly perfect vacuum in the upper part of the tube, except for the slight pressure of the mercury vapor, the instrument indicates the absolute pressure of the atmosphere.
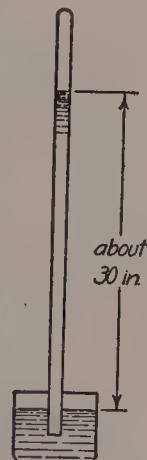
In the real instrument, the surface of the mercury in the cistern is adjusted to a constant level. An ivory pointer is provided in the cistern and the surface of the mercury is always brought up to this point, by means of an adjusting screw, before the instrument is read. A vernier is provided on the scale for fine readings.

Correction of Mercury Barometers. When using a mercury barometer the following should be observed:

Courtesy
Taylor In-
strument Co.

FIG. 16.
Standard
Mercury
Barometer.

(a) Adjust the level of the mercury in the cistern until the surface just touches the ivory pointer. A piece of white paper held behind the cistern will help, or the observer may watch the pointer and its reflection on the mercury surface and adjust until the pointer and its image just meet.

(b) Adjust the vernier slide until its lower edge is just in line with the meniscus of the mercury column.

(c) Read the vernier and note the temperature indicated by the thermometer attached to the case of the barometer.

Standard barometric pressure is usually taken as 30 in. of mercury at 32° F. and at sea level. The corrections to be applied to the readings of the barometer are for (1) cubical expansion of the mercury and (2) linear expansion of the brass scale. A third correction for the meniscus may also be applied if this correction has not been allowed for in the setting of the scale.

The method of computing the temperature correction follows:

$$R_0 = R \left\{ \frac{1 + B(t_1 - t_2)}{1 + M(t_1 - 32)} \right.$$

where R_0 = the corrected reading;

R = the vernier reading in inches;

B = the linear coefficient of expansion of the brass slide (0.00001043);

M = coefficient of cubical expansion of mercury (0.0001012);

t_1 = temperature, degrees F., at which reading was taken;

t_2 = temperature, degrees F., at which brass scale reads correctly, usually 62° F.

Aneroid Barometers

The aneroid type of barometer employs an evacuated metal drum to give indications of atmospheric pressure. The slight movement of the drum head, occurring with changes in atmospheric pressure, is magnified by a mechanism which moves a pointer over a dial graduated in inches of mercury. This type of instrument, while portable and convenient to use, does not retain its calibration and must be handled carefully. The words VERY DRY, FAIR, CHANGE, etc., often printed on the dial, are meaningless.

Aneroid barometers are most conveniently calibrated by comparison with a standard mercury barometer. A small adjusting screw is provided,

usually at the back of the case, by means of which the hand may be made to indicate correctly in the usual working range. Considerable variation in atmospheric pressure will occur during the day and from day to day, so that a calibration is simply a matter of time and comparative readings taken at intervals of a few hours. From such a set of readings a calibration curve may be drawn but it must be remembered that aneroids do not retain their calibrations over long periods and fre-

Courtesy Taylor Instrument Co.

Fig. 17.—Aneroid Barometer.

Courtesy American Paulin System, Inc.

Fig. 18.—Paulin System
Barometer.

quent calibrations are necessary. This is especially true if the aneroid is subjected to any rough handling in the course of ordinary use.

One of the outstanding disadvantages of the usual aneroid barometer is the effect of friction of the internal mechanism which produces a lack of sensitivity in the instrument and also causes the readings, when the barometer is rising, to differ from those obtained when it is falling.

This objection has been overcome in a new type of aneroid known as the *Paulin Barometer*, Fig. 18. In this instrument, the force of atmospheric pressure is balanced by a spring which is always in tension. The tension is changed by a manual adjustment, balance being indicated by a small pointer. Another pointer, attached to the adjusting knob, indicates the barometric reading. It is claimed that this instrument has an accuracy approximating that of a mercury barometer.

5. Calibration of a Bourdon Gage

Principles. The Bourdon gage consists essentially of a hollow circular spring which is deformed when subjected to internal fluid pressure, the deformation causing a pointer to rotate upon a graduated dial, Fig. 19. The pointer is readily removed so that it can be set at any part of the dial to correspond to an applied known pressure.

It should be noted that the deformation of the hollow tube is proportional not to the absolute pressure within the tube, but to the difference
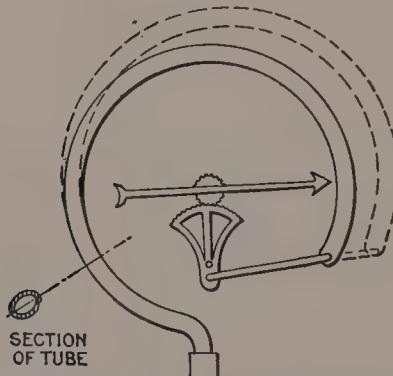
Fig. 19.—Mechanism of Bourdon Gage.
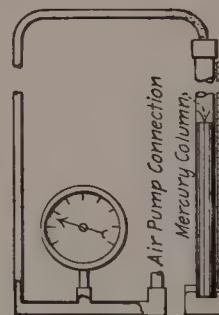
Fig. 20.—Vacuum Gage Tester.

of pressure within and without. The Bourdon pressure gage, therefore, always indicates pressures above atmosphere, since the outer surface of the tube is subjected to atmospheric pressure; and, to convert its readings into absolute pressures, one must add the barometric pressure expressed in the same units.

The vacuum gage works on the same principle as the one just described, the only difference being that the excess of pressure is on the outside of the tube (Fig. 19) causing a contraction of the coil, instead of an expansion, which is indicated, generally, in inches of mercury less than the barometric.

The testing apparatus consists of a chamber in which any desired pressure may be obtained and to which is attached some accurate measuring device. The pressure is generally obtained by some simple form of hand pump. The measuring device is either a "test gage," a column of mercury, or a set of weights acting on a plunger of known area arranged to produce the pressure desired. The test gage is not a desirable standard as it requires calibration from time to time. The mercury column is a

cumbersome and expensive apparatus for pressures above a few pounds, although it is the most accurate method of measuring the true pressure. Besides, a high degree of accuracy is not needed since the least count of commercial gages is generally not less than 5 lb. It is necessary, however, to use the mercury column when testing vacuum gages, and convenient since they are graduated in inches of mercury and not in pounds per square inch (see Fig. 20). The standard weight method is convenient
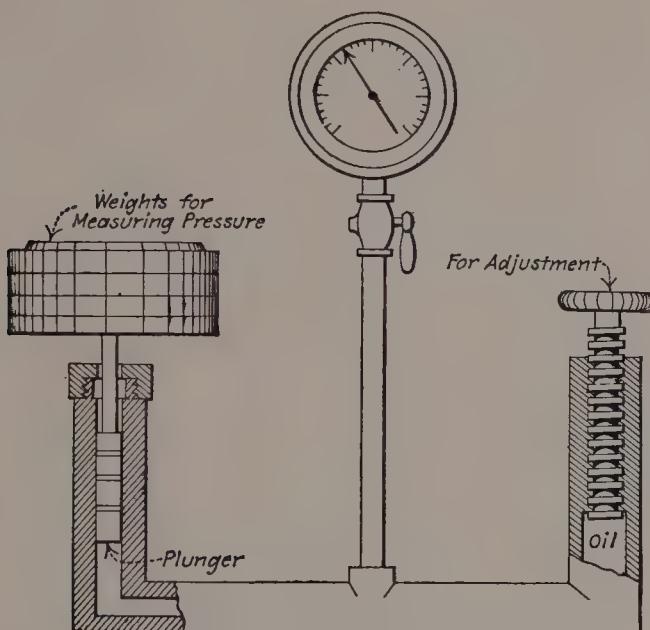
Fig. 21.—Dead Weight Gage Tester.

but the friction of the plunger prevents true records. Fig. 21 shows an apparatus of this type.

(a) **The constants of the standard weight apparatus** are the area of the plunger and the actual weights of the test weights, from which may be figured the actual pressures produced by them. The area is measured by calipering the plunger. The weights should be determined by comparison with standard weights with a degree of precision compatible with the least count of the gage to be tested. Note that the pressure produced by the plunger and attached pan is always applied.

(b) **Calibration.** Set the pointer to read accurately at the division most used and then take a series of readings of true pressures and instrument readings increasing and decreasing, from which plot a calibration curve. Convenient procedure is as follows. With the weights applied to produce the desired pressure, depress the plunger a trifle by hand and

close the cock between the gage and the pressure chamber, thus confining in the gage a pressure slightly greater than that produced by the weights. The pan and weights are now revolved to reduce friction at the plunger and the cock slowly opened. The pressure indicated by the gage will then slowly decrease and a reading may be taken. For increasing values, the pan is raised a trifle, otherwise the procedure is the same. The two readings thus found are added and divided by two to eliminate the effect
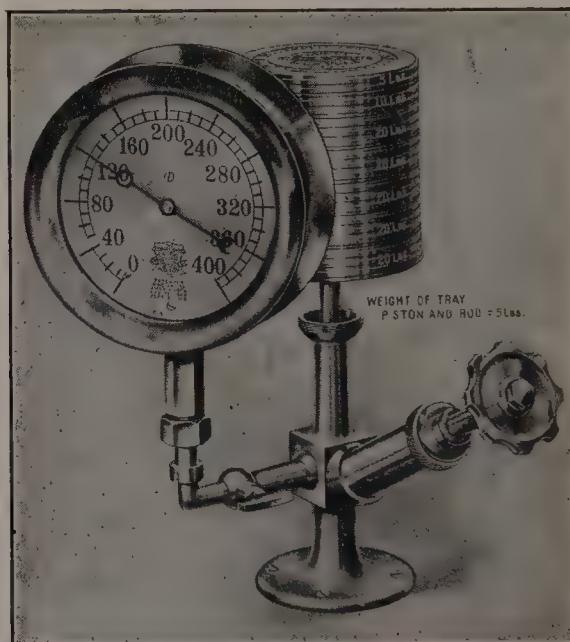
Fig. 22.—Crosby Dead Weight Gage Tester.

of friction. If the gage is to be used for ascending pressures, the calibration resulting from them only should be used. Generally this is not the case and the calibration from the combined readings is preferable.

(c) **Adjustment of the Scale.** When the calibration curve is plotted, if the instrument is in error, it will be noted that the indications either increase or decrease with relation to their true value. Inside of the gage will be found an adjustable link, by means of which the travel of the pointer may be made greater or smaller for a given motion of the tube. This link should be changed in length until a correct motion of the pointer on the scale is found, as proved by a calibration curve.

(d) **The calibration of a vacuum gage** is essentially the same as for a pressure gage, an air pump and mercury column being used instead of the apparatus described.

(e) Calibration of a Recording Pressure Gage. The working principle of the usual pressure recorders is that of the Bourdon tube, the tube being helical in form instead of circular. This provides a sufficient motion to the free end of the tube, to which a pen arm is attached; and magnification by links is not needed. The pen arm swings over a circular chart which is uniformly rotated by clockwork. The curve traced by the pen is thus one of pressure shown radially against time circumferentially (see Fig. 23).

When calibrating, the clock should be stopped, and the instrument read the same as a simple indicating gage. The deadweight apparatus
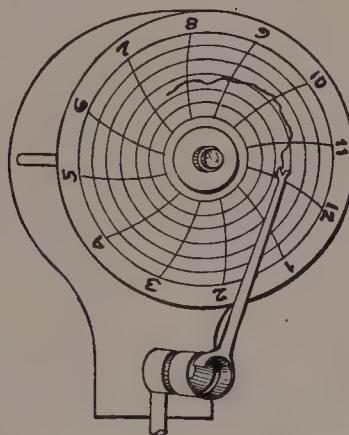
Fig. 23.—Mechanism of Pressure Recorder.

may be used, the pen arm first being set to indicate accurately at the desired point on the chart scale.

Recorders are often set at a considerable distance from the points at which the pressure is to be ascertained. If the gage is either above or below such a point, and the connecting tube is full of liquid (such as condensed steam when the gage is below a steam pipe whose pressure is sought), a correction for the head of liquid must be applied. Since 2.31 ft. head is equivalent to 1 psi., the correction for a difference of height of H feet is

$$H/2.31 \text{ psi.,}$$

and this should be subtracted from the instrument indications when the gage is set below the point of measured pressure, and added when above.

(f) Calibrating High Pressure Gages. The calibration of high pressure gages is essentially the same as described, for ordinary pressure gages, except that a hydrostatic tester is used which eliminates handling

a larger number of heavy weights. (See Fig. 24.) The pressure is applied by means of a hand wheel and readings are taken when the beam is balanced. The other hand wheel is kept in rotation while the readings are being taken, its function being to cause a slight jarring of
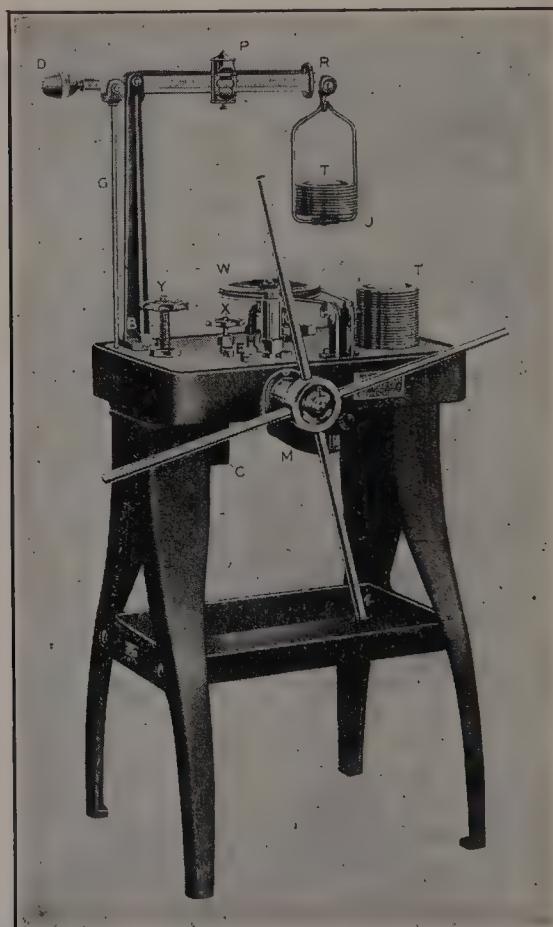
Fig. 24.—Crosby Hydrostatic Tester.

the mechanism which serves to reduce the friction of the moving parts. The device is made in two sizes, 0 to 1500 psi., and 0 to 12,000 psi.

THERMOMETRY

Thermometry is the science of the measurement of temperature. There has been a tendency in the past to confine the definition of thermometry to the measurement of temperatures below 1000° F. and to assign the word *pyrometry* to the measurement of the higher temperatures. There

is no need for this distinction since we now have instruments which cover both ranges in a single device.

Definition of Temperature. Temperature is a measure, on an arbitrary scale, of the intensity of that part of the thermal energy of a body, which is due to the velocity of the molecules. The molecules of a body at a high temperature have greater velocities than those of a body at low temperature. It is evident then, that temperature is a measure of the kinetic energy of the molecules.

By virtue of this property, a body of high temperature will always lose thermal energy to a body of lower temperature. The transfer of thermal energy may take place by means of conduction, convection or radiation, and will continue until the temperature of both bodies is the same.

Thermometers. The expansion of materials, when heated, is approximately proportional to the heat added. It follows that the increase of the length of a material, such as a bar of iron, is a measure of the temperature. Upon this principle, the mercury thermometer was devised, mercury being chosen because of its expansive properties. The Fahrenheit scale, which is almost universally used in engineering, was chosen in a roundabout manner by its inventor, who exposed his thermometer first to freezing water and then to water at the boiling point, under atmospheric pressure in both cases. He marked the stem of the instrument at the level of the mercury in each case, and divided the distance between the marks into 180 equal parts. He continued the scale below the freezing point of water by 32 of these divisions. The freezing point thus became 32°, and the boiling point 212°. Therefore, the Fahrenheit degree may be defined roughly as the increase of temperature corresponding to $\frac{1}{180}$ of the total linear expansion of the material chosen for a thermometer between the freezing and boiling points of water under condition of standard atmospheric pressure.

The material may be liquid, solid or gaseous. But no known material expands uniformly in proportion to the heat added. The departure from uniformity, although slight, is appreciable in precise work, especially at high temperatures. It follows that the divisions of a mercury thermometer, although equal, do not measure equal amounts of heat, and a degree of temperature is a quantity varying with its location on the scale. Also different materials expand according to different laws. Consequently, two thermometers of different materials, standing the same as the freezing and boiling points of water, will differ at all other points. It must not be thought that this is because either one is inaccurate; it is simply

that they are different standards. The linear expansion of a particular material, such as mercury, yields an entirely definite, although arbitrary, temperature scale, because the expansive properties of such a material, when pure, are constant.

In a mercurial thermometer, however, the degree is not measured by the expansion of mercury only; the glass containing it also expands. The scale, therefore, depends upon the relative expansion of mercury and glass. The latter varies considerably both in manufacture and expansive properties, so that for a standard scale it must be specified with great care.

Standard Temperature Scales. In 1887, the International Bureau of Weights and Measures adopted the constant-volume gas thermometer as the standard temperature scale. This is a centigrade scale having two fixed points: the boiling point of pure water under an atmospheric pressure of 760 mm. of mercury and the freezing point of pure water under the same pressure. For the lower range, up to about 450° C., the constant volume hydrogen thermometer is used and the initial pressure of the hydrogen is 1000 mm. of mercury. For the higher range, from 450° C. to 1100° C. the constant-volume nitrogen thermometer is used to avoid the chemical activity of the hydrogen and its tendency to permeate the glass walls of the envelope at elevated temperatures.

Constant-volume gas thermometers are founded upon Boyle's Law; for a perfect gas at constant volume, the pressure varies directly as the temperature. The gas thermometer requires considerable skill and experience in its operation and, therefore, is not used in engineering work. Other secondary standards have been devised which are more satisfactory for ordinary use. The standards, adopted by the National Bureau of Standards, Washington, D. C., can be found by reference to several circulars and bulletins published by that agency.

6. Calibration of High-Reading Thermometers and Pyrometers

Mercury-in-Glass Thermometers are made for measuring temperatures up to about 550° F. They are also made suitable for temperatures up to 1000° F. by filling the capillary tube with nitrogen gas. By making the thermometer of fused quartz and filling the capillary, above the mercury, with nitrogen or carbon dioxide it may be used for temperatures up to 1500° F.; however, quartz thermometers are too expensive for ordinary use.

The use of mercury-in-glass thermometers is somewhat limited because of their fragility and the fact that the sensitive bulb and the graduated scale must necessarily be very close together.

The most convenient and best type of mercury and glass thermometers for engineering test purposes are those having the graduations etched or
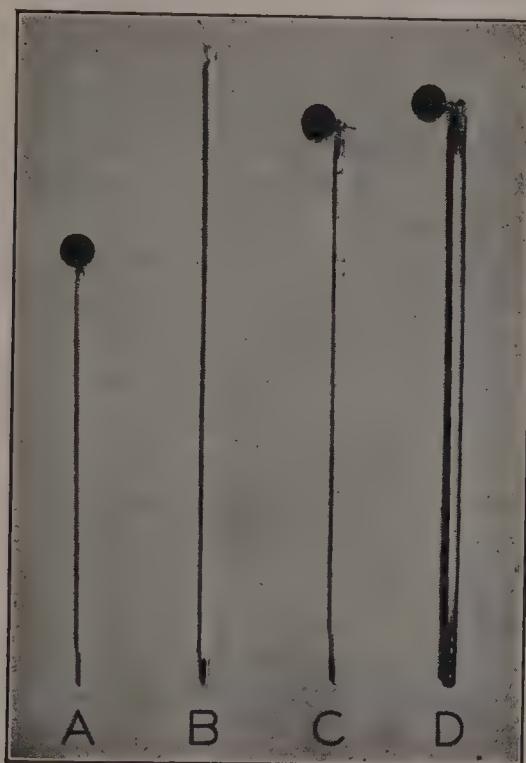
A. Low range. B. Calorimetric. C. High range—nitrogen filled. D. Armored.

FIG. 25.—Commercial Thermometers.

cut in the stem and filled with ink or paint. They may be had in a variety of ranges and either plain or armored. Special thermometers may be obtained having graduations to 0.01° for use in accurate work. The following table gives the ranges and probable accuracy of good grade commercial mercury-in-glass thermometers:

Construction	Accuracy, °F.	Range, °F.
Ordinary lead glass, mercury filled..................	1 to 2	−38 to 575
Jena or Corning glass, mercury and nitrogen filled.....	2 to 10	−38 to 1000
Fused quartz, mercury and nitrogen filled...........	2 to 10	−38 to 1500

Fluids, other than mercury, used for filling glass thermometers are:

Fluid	Freezing Point, °F.	Boiling Point, °F.
Ethyl alcohol...........................	−170	+173
Toluene...............................	−134	+231
Pentane..............................	−203	+ 97

It will be noticed that these fluids are more suited for thermometers used to measure low temperatures.

A special type of mercury-in-glass thermometer is the Beckmann differential thermometer used in accurate calorimetric determinations. It is unique in that its range can be shifted. A chamber is provided at the top to accommodate any mercury which is not required; and by expelling the excess mercury into this chamber, the zero of the scale may be made to correspond to any desired temperature over a wide range. The scale is usually 5° C. long and is arbitrarily numbered from 0 to 5. By use of a low-power reading glass it is possible to read a Beckmann thermometer to 0.001° C.

This type of thermometer has a large time lag and, in calorimetric work, it is necessary to make allowance for the heat capacity of the immersed portion.

Fig. 25 shows a variety of types of commercial thermometers.

Recording thermometers with disc charts have found much favor in power-plant work because of their convenience and accuracy. These are of the constant-volume type, working, in reality, as a pressure gage. The helical tube of the gage is connected by means of a flexible tube of small internal diameter to a bulb, and the system filled with a thermometric medium. Upon heating the bulb, the pressure of the medium is raised, transmitted to the helical tube, and recorded in terms of temperature. The working medium usually is a liquid for low temperatures, a vapor for medium, and a perfect gas for high. The first and the third employ charts with uniform scales, but the graduations of vapor (as alcohol) thermometers increase with the range. These are affected by changes of temperature of the flexible and Bourdon tubes. Such changes cause error with the thermometers using liquids and gases, unless equipped with special compensators.

The table below shows the probable accuracy, range and thermometric media of commercial recording thermometers:

Type	Medium	Accuracy, °F.	Range, °F.
Vapor pressure......	Alcohol	2 to 10	200 to 400
	Ether	2 to 10	100 to 300
	Sulfur dioxide	2 to 10	20 to 250
Liquid filled.........	Alcohol	2 to 10	− 50 to + 300
	Mercury	2 to 10	− 38 to +1000
Gas filled...........	Nitrogen	2 to 10	−200 to +1000

Another form of **expansion thermometer** takes advantage of the **relative expansion** of two metals. As they are heated, the motion of one relative to the other is magnified by a gear combination and transmitted so as to rotate a needle over a graduated dial, thus indicating the temperature.

This type of thermometer has a range from 300° to 1000° F., but the accuracy is very uncertain. The better makes have a probable accuracy of 5° F. for the medium ranges.

The thermoelectric couple is one of the most accurate and convenient instruments for measuring temperature. It depends upon the fact that two wires of different metals having appropriate thermoelectrical properties will generate an electromotive force when connected at their ends, which electromotive force is proportional to the temperature difference between the ends. If it is arranged that the potential be measured with a sensitive millivoltmeter, placed at any convenient distance from the thermoelectric "couple," we have an indirect measure of temperature. The millivoltmeter may be graduated in degrees directly or a calibration curve or conversion table may be used to translate the electrical units into degrees. For more precise work, the potential is measured by means of a potentiometer wherein the e.m.f. of the thermocouple is balanced against the e.m.f. of a standard cell.

The accuracy of measurement depends very largely upon the sensitivity of the indicating instrument. Where the e.m.f. of the couple is measured by a millivoltmeter, the accuracy is from 3° to 20° F. On the other hand, if the e.m.f. of the couple is balanced against a standard cell of known e.m.f., by means of a potentiometer, the accuracy can be made much greater; for the usual laboratory model potentiometer it is about

0.5° F., whereas precision-type potentiometers can give readings as close as 0.02° F.

For temperatures up to 900° F., the couple is composed of a wire of copper and another of constantin. Iron and constantin couples are used for temperatures up to 1500° F.; while for temperatures up to 2800° F., a noble-metal couple is used, composed of one wire of platinum and the other of an alloy of platinum with 10 per cent rhodium. These are by no means all the possible combinations, but they are the ones most commonly used. Manufacturers will supply wire, in various sizes, which has been carefully standardized. They will also supply tables for translating millivolts to degrees.

It should be noted that the connection of the wires to the indicating instrument will also constitute a thermocouple. This fact will introduce an error unless precautions are taken to eliminate this effect or correct for it. In cases where a reference junction is used, such as a bath of melting ice, the wires connected to the indicating instrument will be of the same kind. Thus there will be two opposing couples at the connections which will cancel if the binding-posts are both at the same temperature.

If the binding-posts are used as the cold junction, then a correction must be made to determine the temperature. If the temperature of the "cold" junction is known by placing an ordinary thermometer near the binding-posts, the correction is made by adding the millivolts, corresponding to the junction temperature, to the indicated millivolts. The sum is then converted to degrees. For temperatures lower than the junction temperature, the difference is taken.

This correction is not strictly true because the two couples at the binding-posts are not the same and, therefore, do not cancel out completely. However, such a connection is never used in precise work and the error is negligible for ordinary work. Some indicating devices are equipped with compensators by which the correction can be set into the instrument thus saving the trouble of having to correct each time a reading is taken.

The electrical resistance thermometer is one of the most accurate of temperature-measuring devices. This consists of a length of wire material, capable of resisting the heat, as platinum or nickel, the resistance of which can be measured with a Wheatstone bridge. When the temperature of the wire increases, its resistance goes up, so that the one can be measured by the other, when the coefficient of resistance is known (that is, the change of resistance per degree of temperature). This coefficient

may be determined with great accuracy, and from it can be plotted a calibration curve of temperatures against resistance.

As was the case with the thermocouple type of thermometer, the accuracy of the resistance type of thermometer depends upon the indicating instrument. The range of the resistance thermometer is quite large, from −300° F. to 1800° F., whereas the accuracy may be from 0.001° to 10°, depending upon the equipment used.

Optical pyrometers are advantageous when the hot body to be measured is inaccessible to a couple or bulb. One of them depends upon a comparison of the brightness of the hot body with that of an electric filament which is caused to glow more or less brightly by varying the current passing through it. Then, by measuring the current, an indirect measure of the temperature is obtained as with the thermoelectric couple. When used to measure furnace temperatures in boiler testing, a thin plate of iron is hung at the required point and its brightness is compared with that of the filament. This type of pyrometer is not very precise because it is difficult to judge exact similarity of brightness.

Another optical pyrometer depends upon the thermoelectric couple principle, but the couple is acted upon by radiant heat concentrated upon it by means of lenses in the form of a telescope.

There are other forms of pyrometers using other principles, but those listed above are the principal ones.

There are three ways generally used to calibrate high-reading thermometers and pyrometers: first, by comparison with a standard or secondary standard instrument; second, by comparison with the temperature of saturated or wet steam determined from its pressure; third, by comparison with temperatures as shown by the melting-points of metals or boiling points of fluids and salts.

(a) **Calibration against a Standard.** For this purpose, mercury thermometers may be used for temperatures up to 1000° F. Above, and often below that temperature, a favorite instrument is the thermoelectric couple, as it is one of the most accurate and easy to use.

The temperature is varied in a gas or electric furnace in which the bulbs of the standard instrument * and the one to be tested are placed. Care should be taken to bring the whole furnace up to a uniform temperature before recording observations, by allowing sufficient time for heating. The sensitive portions of the two instruments should be placed as close to each other as possible to avoid errors from localized tempera-

* A standard thermometer is one which has been calibrated by the Bureau of Standards, Washington, D. C.

tures. It is best to take a series of decreasing readings to supplement the increasing ones, allowing the furnace to cool for this purpose. If the up and down readings are different at a given temperature, as they will be with certain types of instruments, they should be averaged.

(b) Calibration against Steam Temperatures. This method is limited by the available pressure. At 200 lb. gage, the temperature is 388° F.,

Courtesy The Bristol Co.

Fig. 26.—Recording Thermometer.

and at 500 lb., about 470° F. It is thus apparent that not very high temperatures can be reached with saturated steam.

The procedure is to vary the temperature in a steam chamber similar to that used for testing indicator springs, Fig. 61, p. 98, by controlling the pressure. The thermometer is inserted in a well set in the chamber and readings taken of it and of an accurate and sufficiently precise pressure gage at each variation of the pressure. The actual temperatures are ascertained by reference to the steam tables. A barometer reading is necessary to get absolute pressures.

When the steam is throttled to reduce its pressure it may become super-

heated, in which case its temperature is not determinable. To avoid this, water may be sprayed into the steam before it enters the steam chamber, or the chamber may be water-jacketed. It is essential that there be definite knowledge of the steam's wetness, for which purpose a throttling calorimeter (see Test 7) may be referred to.

A calibration of an instrument indicating beyond the available temperatures may be obtained by extrapolation. That is, after the experimental data have been plotted, the curve is continued beyond the known points. If the curve is a straight line, the result may be good, but caution is necessary not to extend the extrapolation too far.

(c) Calibration against Boiling Points. This method will be found convenient over quite a large range. The accuracy depends upon the purity of the materials.

Liquids and salts, when used, should be boiled in the standard Meyer tube form of boiling-point apparatus, Fig. 27, which can be heated over a gas flame. The asbestos or aluminum cone, placed around the bulb of the thermometer, prevents condensed vapor from reaching the bulb and cooling it below the temperature of the surrounding vapor. The cone also

FIG. 27.—Standard Meyer Tube Form of Boiling Point Apparatus.

prevents direct radiation from the bulb to the colder walls of the Meyer tube.

When calibrating thermometers against substances of known boiling points, the thermometers are immersed in the vapors of the boiling substances. The materials most commonly used with their boiling points are given in the following table:

Substance	Boiling Point, °F.
Naphthalene.............	424
Benzophenone............	582
Anthracene..............	644
Sulfur..................	832

The boiling-point temperatures are for standard barometric pressure (29.92 in.). For any other barometric pressure a correction must be applied.
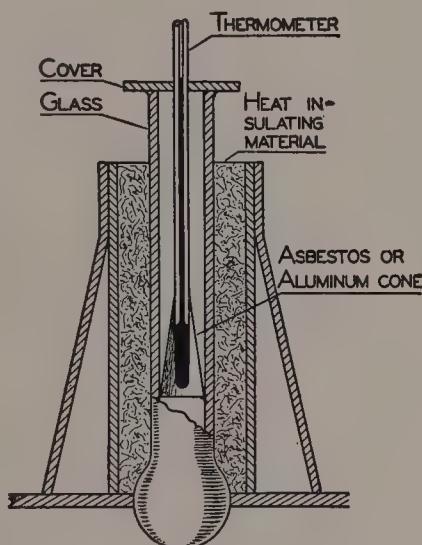
$$T = t_0 + A(p - 29.92)$$

where T = the boiling point at pressure p;

t_0 = the boiling point at 29.92 in.;

A = correction factor in following table.

Substance	Boiling Point Correction Factor A
Naphthalene...............	2.61
Benzophenone.............	2.88
Anthracene...............	3.(2)
Sulfur.....................	4.21

For an extended discussion of the calibration of mercury-in-glass thermometers and other temperature-measuring devices, the reader is referred to Chapter 3, Test Code on Instruments and Apparatus, published by the American Society of Mechanical Engineers, and also to numerous bulletins on the subject published by the Bureau of Standards, Washington, D. C.

(d) Installation and Use of Thermometers. There is a popular idea that the measurement of temperature is one of the most accurate measurements performed in engineering tests. The true state of affairs is quite to the contrary, accurate temperatures being quite impossible of attainment, in some cases, under the present state of the art.*

A very large source of error lies in the natural tendency toward temperature equalization, the tendency which causes the flow of heat from regions of high temperature toward regions of lower temperature. As an example of this effect, assume a thermometer well inserted into a pipe carrying hot gas or steam. The pipe is exposed to outside air and is probably cooler than the gas or steam which it is carrying. The thermometer well is attached to the pipe and, while probably not the same as that of the pipe, its temperature is lower than the medium in the pipe since heat tends to flow along the well toward the cooler outside region. A thermometer placed in the well will indicate a temperature lower than the true temperature of the medium.

Quite often a thermometer or thermocouple must be installed in such a manner that it can "see" surfaces of a temperature considerably higher

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* The Bureau of Standards has developed a method of obtaining true temperatures. The method consists of simultaneously measuring the temperature of a hot body with thermocouples of different bead diameters. The indicated temperatures are then plotted against bead diameter. Extrapolation of the resulting curve to zero bead diameter is assumed to give the true temperature.

or lower than that of the medium in which the temperature-measuring device is immersed. The thermometer bulb or thermocouple, in such a case, will of course gain heat from, or lose heat to, such surfaces by radiation. The effect of radiation may be, to a large extent, eliminated by the proper use of suitable shielding. In most cases a cylindrical shield is used with the axis parallel to the flow of the medium whose temperature is to be measured. Individual installations should be studied carefully to make sure that the shield produces the desired effect since it is possible for such shielding to create errors greater than those it is intended to eliminate.

Another source of error is in the failure to consider the mass flow of the medium whose temperature is being measured; the medium may be in a stratified condition in the duct and not only are the strata likely to vary in temperature, one from the other, but are also likely to be moving with different velocities. Furthermore, the rate of flow of the whole mass may be variable and, unless the temperature were quite constant, readings taken at equal intervals of time would not give a true average.

(e) Thermometer Wells. Wherever possible the bulb of the thermometer should be exposed directly to the body or medium whose temperature is to be measured. When this is not possible a thermometer well should be installed.

There are two general types of wells; plain walled for use with liquids and saturated vapor and fin-walled for use with gas and superheated vapors. The fins are provided in the latter case to increase the rate of heat transfer. Wells of both types should be made with their parts, which project beyond the container, as small as possible to reduce the flow of heat to or from the surrounding region. If the container is insulated, the insulation should be carried well around the mouth of the thermometer well.

Thermometer wells should be filled with some sort of liquid suitable to the conditions of use. For low temperature and where the temperature variation is not great, lubricating oil may be used. The great fault with oil is its low thermal capacity which tends to cause a time lag if there is much variation in the temperature. For higher temperatures, up to 500° F., mercury may be used in iron-walled wells. It should never be used in brass-walled wells. Solder and block-tin may be used for higher temperatures but the thermometer should be removed before the metal in the well cools, otherwise it will be crushed by the contracting metal.

Wells should be deep enough to prevent an excessive amount of emergent thread of mercury. The thermometer should not be lifted from the

well when being read. Furthermore, care should be exercised to have the line of sight perpendicular to the stem of the thermometer in order to avoid the effect of parallax.

(f) Stem Corrections. If a mercury thermometer is to indicate correctly, all of the mercury, both in the stem and bulb, should be subjected to the measured temperature. In practice this is rarely the case, as part of the stem containing mercury is at or near room temperature. For accurate work it is then necessary to apply a "stem correction."

Let S = stem correction in degrees;

H = height of column above actual position of mercury, if uniformly heated, inches;

l = length of one degree on the stem, inches;

N = number of degrees on the stem exposed to room temperature;

T = temperature, Fahrenheit, of the bulb;

t = average temperature, Fahrenheit, of the exposed mercury and stem;

C = cubical coefficient of expansion of the glass and mercury, combined by subtracting the value of one from that of the other.

Then

$$H = C(l \times N)(T - t)$$

Also

$$S = \frac{H}{l}.$$

Hence

$$S = CN(T - t)$$

The value of C for the Fahrenheit scale may be taken as 0.000088 and for the centigrade scale, 0.00016.

It is generally sufficiently accurate to take the actual reading as the value of T. If desired, the corrected temperature thus obtained may then be used as T, and a second and more nearly exact stem correction calculated.

The value of t is somewhere between room temperature and that of the bulb. It is often estimated by hanging a second thermometer against the one to be corrected, with its bulb at a middle point on the exposed mercury column. It is very doubtful that this yields a close value for t.

Whether or not a stem correction is necessary depends upon the purpose for which the temperature is measured. In engineering work, thermometers are generally used to measure temperature differences, so that the percentage of error in the result may be based upon a much smaller

quantity than the actual number of degrees read. In many cases, when the temperatures are comparatively low or when the stem is well immersed, the correction is negligible.

ENTHALPY OF STEAM—CALORIMETRY—SAMPLING

The accepted unit of enthalpy in American engineering practice is the "mean British thermal unit," abbreviated B.t.u. This is one one-hundred and eightieth ($\frac{1}{180}$) part of the total amount of heat necessary to raise the temperature of one pound of water from its freezing point, 32° F., to its boiling point, 212° F., at an absolute pressure of 14.696 psi.

The mechanical equivalent of the B.t.u. is, very nearly, 778 ft.-lb.

In this work, the symbol H_2O will be used as a generic expression to represent water, steam, a mixture of steam and water, or superheated steam. According to modern chemistry, water or ice may have a much more complicated structure than is indicated by the simple form, H_2O, but science has not assigned a name to cover the material, water, in its various states. Such a name is sorely needed in engineering terminology to insure clarity of meaning; in the absence of a generally accepted term, H_2O will be used as the best one available.

H_2O, as a working medium in heat-power engineering, must be considered in the following states, in each of which it follows different physical laws: (1) ice; (2) liquid; (3) liquid and vapor mixture; (4) vapor; (5) imperfect gas; (6) perfect gas.

The first state is of importance in mechanical refrigeration and need not be considered in this section.

In all of the following discussion one pound of H_2O, in whatever state, is understood. Enthalpy quantities are for one pound and where the word *volume* is used, the cubic feet of space occupied by one pound of H_2O is implied. This can also be termed *specific volume*.

The various states of H_2O, from liquid to perfect gas, are definitely established by the simultaneous properties of pressure and volume. These properties necessarily are in equilibrium at certain definite temperatures. In any state, there is a definite enthalpy, intrinsic to the H_2O, when counted above that at a datum state. The datum state is conveniently taken as water at 32° F.

The word *enthalpy*, which has been adopted into engineering literature of late years, is defined as the total amount of heat energy, in B.t.u., necessary to be added to one pound of water at 32° F. to change to a given state, the pressure on the H_2O being constant.

The phenomena accompanying the increase of the enthalpy of water, initially at 32° F., are as follows. Assuming constant pressure on the H_2O, the temperature of the water rises and its volume, at first, decreases until the temperature reaches 39.2° F. after which the volume increases until the temperature attains the value necessary for the water to boil at the given pressure. There is then a large increase of volume, without increase of temperature, and heat energy is absorbed to change molecular positions from those in the liquid to the vapor state. This goes on until all the water is evaporated; if no more heat energy is added after this stage, the H_2O is said to be *saturated*.

Internal Energy. At this stage, the steam contains energy in two forms: first, as indicated by its temperature; and, second, as *potential* energy by virtue of the increased distance between its molecules in the vapor form as compared with the liquid form, sometimes called the "internal latent heat."

A further addition of heat energy increases the temperature and volume of the steam. This produces the phenomenon known as *superheating*.

Total and Partial Enthalpy. The *heat added* during the constant pressure process equals the sum of all the energy changes occurring in the H_2O. These are the increase of internal energy as indicated by the rise in temperature, the internal work required to increase the molecular distances, and the external work necessary to make room for the increased volume of the vapor. This quantity is of prime importance to the engineer and is the *total enthalpy* of the vapor; usually spoken of as simply enthalpy.

That part of the total enthalpy required to raise the temperature of the water from 32° F. to the boiling point at the given pressure is called the *enthalpy of the liquid*. The part of the enthalpy which represents the change from the liquid to the vapor stage is the *enthalpy of vaporization*. If superheating is present there will also be the *enthalpy of superheat*. The symbols customarily used for these quantities are:

H_f = enthalpy of the liquid;
H_{fg} = enthalpy of vaporization;
H_g = total enthalpy regardless of whether the steam is saturated or
      superheated.

Enthalpy of Low-Pressure Superheated Steam. Professor Diederichs has shown that the enthalpy of low-pressure superheated steam is independent of the pressure and is a function of the temperature only. He has proposed the following empirical equation:

$$H_g = 1058.7 + 0.455T$$

in which T is the temperature of the steam in degrees F. The coefficient, 0.455, is empirical and is not to be considered as a specific heat. This equation gives values closely in agreement with those of the steam tables, even when T is very near the saturation temperature. The relation is useful in certain heat calculations for which it is impossible to measure the steam pressure directly, and laborious to calculate it.

Steam calorimeters are instruments for measuring the quality of wet steam; determining the amount of moisture present in the mixture of water and vapor. Various forms are made, depending upon different principles of operation. These depend either upon a transfer of heat by which it can be equated to a measurable quantity or upon the mechanical separation of the entrained moisture. The first is strictly a calorimetric process. Thus, if a sample of the steam, whose wetness is to be measured, is condensed in water, the heat given up to the water is readily measured and this quantity can be equated to $H_f + xH_{fg}$ in which x, the quality, is the unknown. Solution of the equation then yields the value of x. Another method is to superheat the steam, in which form its enthalpy is readily determined from the steam tables.

FIG. 28.—Stott and Pigott Sampling Tube.

The mechanical separation method uses an apparatus similar to the ordinary steam separator, and involves weighing both the steam and the separated water.

Sampling. As usually arranged only a small sample of the total steam is tested for a quality determination. The accuracy of the result is primarily dependent upon the representativeness of the sample, so that every care should be used to secure a correct one. Unfortunately, it is always uncertain that a reasonably accurate sample has been obtained, but by using the proper shape of sampling pipe, inaccuracy may be reduced to a minimum.

In a horizontal steam pipe, water is apt to run along at the bottom, separated from the steam. If all of this water is passed into the sampling pipe, an undue amount of water appears in the sample. In a vertical

pipe, moisture runs down the inside of the pipe wall. Undoubtedly some of this should be included in the sample, but it is no easy matter, theoretically or practically, to secure the right proportions. Fig. 28 shows an approved form of sampling tube.

7. THE THROTTLING CALORIMETER

Principles. Fig. 29 represents the instrument diagrammatically. Steam from a sampling tube enters the steam pipe from which it passes through the orifice O, about $\frac{1}{16}$ in. in diameter, into the chamber C, which is open to the atmosphere. The steam is throttled upon passing through the orifice, and drops to a pressure only a little in excess of atmospheric. Now the heat contained by 1 lb. of dry steam at high pressure is greater than that at low pressure. Upon reaching the chamber, C, the steam therefore may contain an amount of heat in excess of that necessary for saturation, and this excess goes to evaporate the moisture carried in with the steam and to superheat both. In this condition, the heat content is readily measured, which makes possible a heat equation involving one unknown. Before entering the orifice, the enthalpy is $H_{f_1} + xH_{fg_1}$. In the calorimeter chamber it is $H_{f_2} + H_{fg_2} + C_p(T_2 - t_2)$. The subscripts 1 and 2 refer to the condition before and after the steam passes through the orifice. C_p is the specific heat at constant pressure of the superheated steam usually taken as 0.48 which is only approximate since the value of C_p varies with pressure and temperature. T_2 is the temperature obtained from the thermometer in the calorimeter chamber while t_2 is the saturation temperature corresponding to the pressure in the calorimeter chamber.

FIG. 29.—Throttling Calorimeter.

These calculations may be carried out graphically on the Mollier Chart if it be recalled that the throttling process is a constant enthalpy process; represented by a horizontal line on the chart. First, find the point on the chart which represents the condition of the steam in the calorimeter chamber; pressure and temperature define this point. Second, proceed horizontally until the line cuts the pressure on the high-pressure side of the orifice. The quality can now be estimated from the moisture lines on the chart.
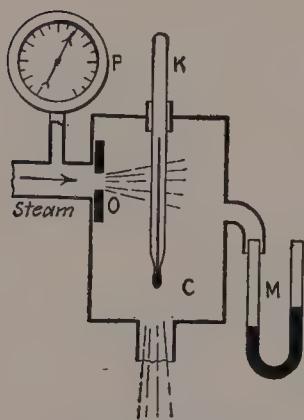
Errors of the throttling calorimeter are due mainly to false thermometer readings of the superheat, and to radiation from the instrument or fittings. It is best to set the thermometer in the calorimeter without using a well, which may be done readily with a perforated plug of rubber, cork or wood. The thermometer should be moved vertically in the calorimeter until the hottest part is found when the steam is passing as in regular use. To avoid radiation, all parts should be well lagged. Not too large an orifice should be used as this raises the back pressure in the calorimeter, and passes more steam than necessary.

A temperature, in the chamber, close to 212° F. is an indication that the calorimeter is not functioning properly either due to excessive moisture in the steam, or water from the walls of the main steam line getting into the sampling tube giving a false sample. Occasional slugs of water will give a temporary depression of the temperature and are an indication that the sample is not very good and the tube should be altered.

Fig. 30.—Jacketed Calorimeter.

In a comparison of indications of various types of throttling calorimeter, set up a number of calorimeters of different construction and heat protection so that they will all receive steam of the same quality. Then compare the temperatures of superheat, and the qualities of steam as shown by each.

Fig. 30 shows a "jacketed" calorimeter designed to avoid error from radiation of heat. The main orifice is at A. Steam also enters the annular space D through another orifice B, thus keeping the walls of the calorimeter chamber C at the same temperature as the steam inside. An advantage of this type is that there is no back pressure on the calorimeter chamber and therefore no manometer need be used.

This instrument may be used as a standard and the errors of other calorimeters, lagged or unlagged, determined by comparison of their indications with those of the jacketed calorimeter.

The equation of the throttling calorimeter is made on the assumption that the heat content of the steam entering is the same as that in the calorimeter chamber. The effect of radiation of heat is to make this assumption untrue, and the thermometer indicating the temperature of the steam in the calorimeter chamber will read lower than if there were no heat radiation from the calorimeter chamber.
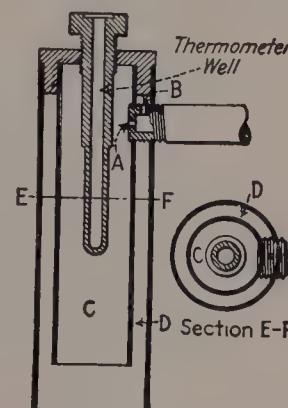
8. The Separating Calorimeter

Principles. If the water entrained in steam is mechanically separated, and the weights of the dry steam and separated water are then measured the percentage of water may be readily calculated. Mechanical separation may be effected by deflecting the steam and water through a sharp bend; the water is then thrown out by centrifugal force.

Fig. 31 shows diagrammatically the Carpenter separating calorimeter. The water collects in the calibrated chamber C and its amount is measured by the scale S placed against a gage glass. The weight of dry steam is measured by the orifice method according to Napier's rule, the orifice being located at O. Since the weight of steam discharged from an orifice into the atmosphere equals a constant times the absolute pressure, an ordinary pressure gage G may be calibrated to indicate the weight of steam passing per minute. In operation, a steam sample is passed through the calorimeter, and, after it is thoroughly heated, initial and final readings of the water level are made for a measured interval of time, together with a number of readings of the gage from which an average is calculated. The gage gives the pounds of dry steam passing per minute; multiplying this by the number of minutes gives the total weight of steam. This weight is then divided by the sum of the weights of water and steam to get the quality.

FIG. 31.—Separating Calorimeter.

The chief advantage of the separating over the throttling calorimeter is that it is operative no matter how low the quality. Experiments have shown that complete separation of the water may be depended upon with a properly designed separator.

(a) **Calibration of the dry steam meter** may be made as for Test 30 (a). A calibration curve for the gage should be plotted.

(b) **Calibration of the Water Gage.** The chamber C, Fig. 31, is allowed to fill with water separated from steam passing as in usual operation. A small amount of this water is then drawn off from the pet cock P into a flask of cold water (to prevent evaporation). The weights of the flask and contents and the heights of the levels in the water gage before and after the operation are carefully measured. A number of such determinations furnish the data for a calibration curve.
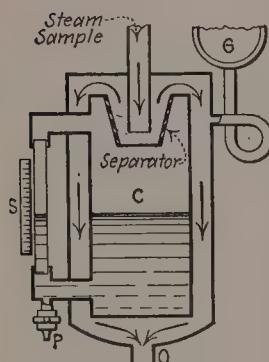
If the water used for the calibration is cold, its weight should be corrected for the difference in density due to the different temperatures. The temperature in usual operation may be taken as that of the steam passing through the calorimeter.   (See Appendix for density of water.)

(c) Determination of Radiation Correction. Radiation from the instrument may cause a greater amount of water to be thrown down than was in the sample. In the Carpenter instrument, on account of the steam jacket arrangement (see Fig. 31) radiation occurs after the steam has left the separator; the water gage then remains correct, but the orifice measurement may be faulty since the orifice passes wet instead of dry steam. If the gage has been calibrated for the particular conditions of radiation, this calibration takes care of the radiation correction.

If radiation causes an increase of water in the water gage the correction may be determined as follows: Dry steam should be supplied to the calorimeter either with an apparatus as described under Test 7(a) or by taking the sample from another steam separator. Any water that shows at the water gage of the calorimeter is then due to radiation. If it is figured as so many pounds per minute, the correction in ordinary use is readily applied.

9. Combined Separating and Throttling Calorimeters

Principles. When dealing with low pressure steam, below about 40 lb. absolute, the usual types of throttling and separating calorimeters are inefficient. In such cases a throttling calorimeter is attached to the discharge of a separating calorimeter. The separating calorimeter must be carefully calibrated for radiation. With such a device, the steam escaping from the separating calorimeter is tested a second time in the throttling calorimeter with the result that it is possible to make fairly accurate determinations on steam of almost any degree of wetness. A special type of sampling tube should be used and the discharge of the throttling calorimeter is attached to a condenser to lower the pressure and increase the capacity of the orifice. A mercury manometer is attached to indicate the pressure on the discharge side.

This apparatus is very satisfactory for moisture determination on exhaust steam lines.

The quality of the steam may be computed, for the combined calorimeter, as follows:

$$x = x_0 \times \frac{W_s + R}{W_w + W_s}$$

where x = quality of steam;

$\quad x_0$ = quality of steam discharge from the separating portion as determined by the throttling portion;

$\quad W_s$ = weight of dry steam condensed after passing through the throttling portion, pounds;

$\quad W_w$ = weight of moisture collected in the separating portion, pounds;

$\quad R$ = correction for radiation, pounds.

W_s and W_w must both be measured for the same length of time.

10. The Condensing Calorimeter

Principles. If a sample of steam, the quality of which is to be determined, is condensed either by mixing with cool water, or by entering condensing coils surrounded by water, the arrangement constitutes a condensing calorimeter.

Let W_s = weight of condensate, including moisture in sample;

$\quad W_w$ = weight of condensing water;

$\quad t_w$ = temperature of water before it is heated;

$\quad T_w$ = temperature of water after it is heated;

$\quad T_s$ = temperature of the condensate.

If condensation is accomplished by mixing, $T_w = T_s$; otherwise they have different values. Now, if the calorimeter operates continuously, and if radiation is neglected

$$W_s[(H_f + xH_{fg}) - (T_s - 32)] = W_w(T_w - t_w)$$

That is, the heat lost by the steam equals that gained by the water.

If the operation is not continuous, the material of the calorimeter absorbs an amount of heat which must be accounted for. This quantity is called the "water equivalent," being the weight of water that absorbs the same amount of heat for a given temperature rise as does the calorimeter. For noncontinuous condensing calorimeters, the water equivalent should be added to W_w.

The equation may be solved for x, the other quantities being obtained experimentally. H_f and H_{fg} are found from the steam tables, the pressure of the steam being observed.

A barrel of water on a platform scales makes a noncontinuous mixing calorimeter. An injector may be used as a continuous mixing calorimeter. A surface condenser may be arranged as a continuous nonmixing calorimeter.

(a) **Determination of the Water Equivalent.** The condenser should be isolated and filled with water either hotter or colder than the material of the calorimeter. The initial and final temperatures of the water, t_1 and t_2, and the weight of water W should be observed. If Q is the water equivalent, and t is the initial temperature of the calorimeter,

$$W(t_1 - t_2) = Q(t_2 - t)$$

since the heat lost by the water equals that gained by the calorimeter, or vice versa. This equation may then be solved for Q.

The equation neglects radiation. To avoid the consequent error, a second test may be made with the temperatures of calorimeter and water so adjusted that the temperature rise up to room temperature approximately equals that above.

For example, suppose that for the first test water at 110° F. cools to 80°, raising the temperature of the calorimeter from that of the room, 70°, to 80°. The rise of temperature of the calorimeter during the test is 10°. Now, if its initial temperature were 65° and its final 75°, radiation to and from the room would be balanced. Hence, for the second test, the calorimeter should be previously cooled to 65°, and then heated with water at 105° so that the 30° drop in the water temperature, noted in the first test, will bring the final temperature to the desired 75°. The variation will not, of course, be exactly equal above and below the room temperature, but by this method the effect of radiation upon the accuracy of the determination of the water equivalent is made negligible.

The student should note the fact that radiation thus taken care of is an entirely different quantity from that which occurs in the usual operation of the instrument. For accurate work, the latter should be taken into account as follows:

(b) **Determination of the Radiation Correction.** For noncontinuous condensing calorimeters such as the barrel calorimeter, radiation correction may be avoided in actual use in exactly the same way as that described for the water equivalent determination. The flow of steam into the calorimeter is continued sufficiently long to raise the temperature of the cold condensing water as much above room temperature as it was below.

For continuous calorimeters, the radiation correction must be found and applied in the equation as follows,

$$W_s[H_f + xH_{fg} - (T_s - 32)] = W_w(T_w - t_w) + R$$
$$= W_w(T_w + t' - t_w),$$

in which R is the heat radiated from W_w lb. of cooling water and $t' = R \div W_w$.

To find t', the calorimeter is operated as in usual practice, except that dry steam is supplied to it either by the method of Test 7(a) or by the use of a steam separator. The value of x is then 1, and the equation may be solved for t'.

The correction applies only to the temperature conditions of the radiation test. If the conditions in ordinary operation are markedly different, the radiation test should be repeated to fit the new conditions. It is possible, however, to keep the initial and final temperatures of the cooling water practically the same for different conditions of the steam samples by varying the rate of cooling water. If this is done, and if the room temperature does not vary much, only a single value of the radiation correction is needed.

ANGULAR VELOCITY

The great majority of machines are either based upon rotary motion or incorporate rotary motion into some part of their mechanism. Consequently, the speed of rotation or angular velocity is a measurement which is encountered practically every time a machine is tested.

The units are usually whole revolutions per minute, abbreviated r.p.m. This simplest form of instrument for the measurement of this quantity is the *hand counter* which consists essentially of a worm and wheel. The worm is part of a small spindle tipped with rubber so that, if held against the center at the end of a revolving shaft, it turns with the shaft, thus imparting rotary motion to the worm-wheel. The worm has a single thread and the wheel has 100 teeth, thus 100 turns of the worm produce 1 turn of the wheel. The wheel carries a circular plate graduated into 100 equal divisions and this indicates the turns of the spindle. In operation, the hand counter is applied to the rotating shaft during a period of time measured by a stop watch or other time piece and, from the readings of the watch and counter, the revolutions per minute are calculated.
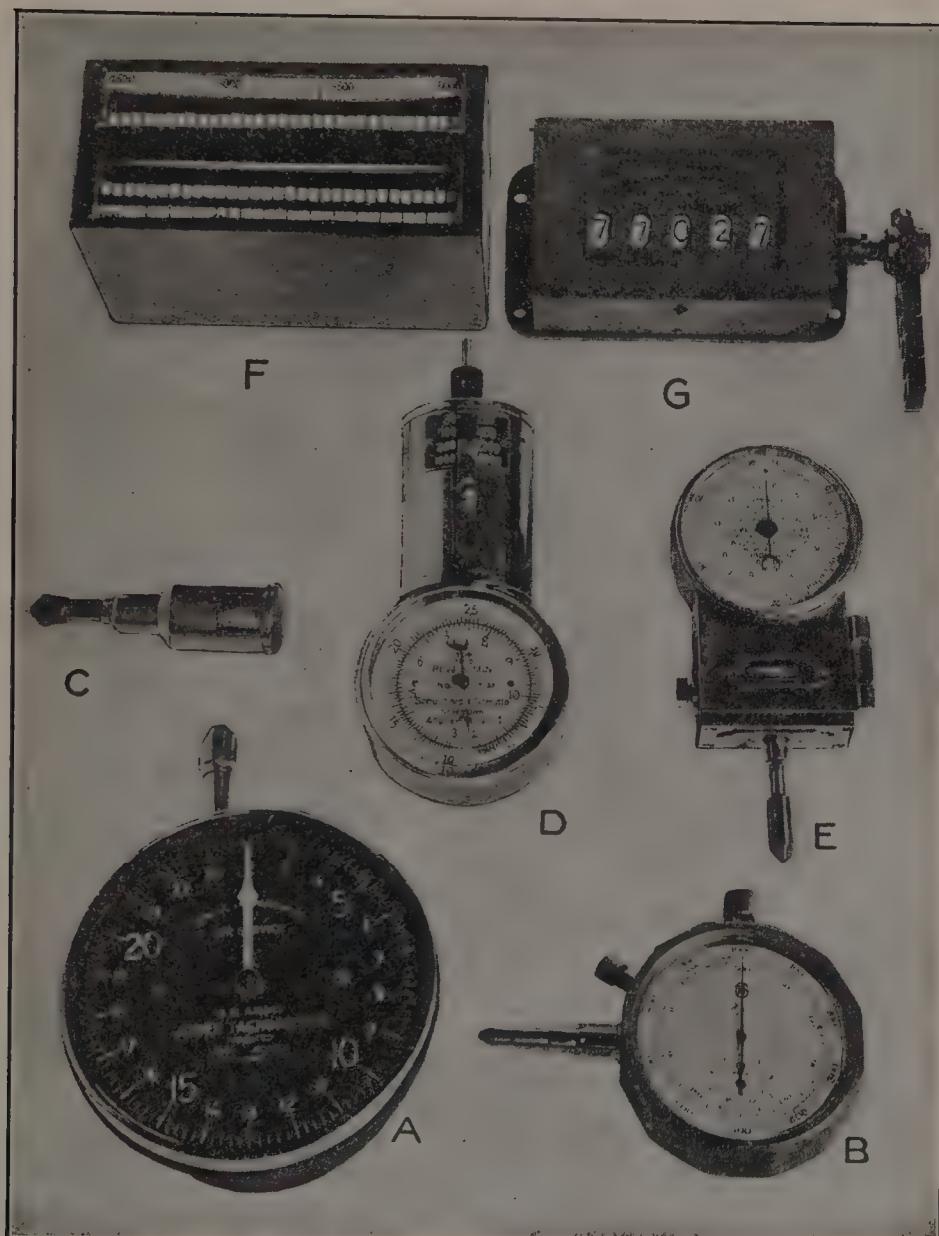
A. Direct Reading Hand Tachometer.
B. Combined Watch and Counter.
C. Hand Counter.

D. } Adjustable Range Direct Reading
E. } Tachometers.
F. Vibrating Reed Tachometer.
G. Continuous Counter.

FIG. 32.—Types of Tachometers and Counters.

Some hand counters have a separate indicator which counts whole turns of the graduated plate. This is quite important in high-speed machinery where the counts of the hundreds of revolutions per minute might easily be wrong due to errors of counting.

A still more refined type of hand counter is illustrated at C in Fig. 32 which incorporates counting wheels. There is also a clutch between the spindle and the counter mechanism which is operated by pushing the body of the counter toward the shaft.

The continuous counter, or cyclometer, is a variation of the hand counter in that its spindle is geared to a series of wheels with numbered faces, partially exposed, so as to show at any time the total number of revolutions. Some forms of continuous counter are driven by a reciprocating lever instead of a revolving spindle. In both forms, the instrument usually receives its motion from a small pin on the end of the shaft whose revolutions are to be measured, which pin acts as a crank.

Several makes of instruments may be obtained which combine the continuous counter and the stop watch in the one case. The instrument is so constructed that both the watch and counter are started simultaneously. In some instruments, the counter hand stops after a predetermined time has elapsed. In others the counter hand revolves continuously but its connection to the stop watch is controlled by a small lever. In either case the timing errors are largely eliminated and this type of instrument is considered the most accurate measure of rotational speed.

The tachometer gives instantaneous indications of revolutions per minute without measuring time. The spindle transmitting the speed bears a pair of weights so linked that they move outward from the spindle by centrifugal force, and against the restraint of a spring. In so doing they actuate a pointer on an appropriately graduated dial. As the centrifugal force, and therefore the motion of the weights, are proportional to the speed, the pointer may register the speed instantaneously.

On account of the variation in their internal friction and in the stiffness of their springs with use, tachometers should be calibrated before using on important work.

The electric tachometer consists of a small magneto which is driven by the shaft whose speed is to be measured. Since the magneto has a permanent field, its output e.m.f. is proportional to the speed. The indications of speed are obtained by a sensitive voltmeter calibrated directly in revolutions per minute instead of volts. When kept in good condition,

this type of instrument is second, in accuracy, only to the combined watch and counter instruments.

The vibrating reed tachometer is of somewhat limited value because of its small range. It has the advantage of requiring no connection to the revolving shaft being simply securely bolted to the machine. They are more desirable for permanent installations.

The tachograph is a very sensitive tachometer, arranged to give an autographic diagram of angular velocity. It is used to measure minute changes of speed within a revolution.

Recording tachometers are made in a variety of forms, and, for the greater part, depend upon the action of centrifugal force on a solid or fluid mass. Thus, the simple tachometer just described may be made as a recorder if the centrifugal weights actuate a pen arm (instead of a dial pointer) which travels over a clock-work driven chart. An example of the centrifugal fluid tachometer is the Bristol. This consists of a small air blower which is driven by the shaft whose speed is to be found. The blower creates a partial vacuum which increases as the speed increases, and vice versa. The vacuum is transmitted to and recorded by a pressure recorder, the chart of which is graduated in revolutions per minute. Still another recording tachometer consists of a small generator, driven by the shaft considered, whose speed variations are evidenced by the generator voltage. This is recorded in terms of revolutions per minute upon a time chart.

11. Calibration of a Tachometer

Principles. Since the advent of the synchronous electric clock, the frequency of most central-station systems is held very close to the standard frequency of the system, usually 60 cycles per sec. but 50 cycles per sec. in some cases. The frequency regulation is so good that the electric clocks seldom are in error more than a few seconds in any 24-hr. period. As a result, we have an excellent standard to which various angular velocity-measuring instruments may be compared by the use of a synchronous motor. This need not be larger than $\frac{1}{8}$ or $\frac{1}{6}$ hp. A number of parallel shafts may be driven from the motor-shaft by means of small spur gears which may be purchased from stock. Such a device is illustrated in Fig. 33 and is by far the simplest and most practical device for the testing of tachometers of all kinds.

Lacking such a device, any rotating shaft whose speed of rotation can be varied may be used but the method becomes more laborious because

it becomes necessary to compare the instrument being calibrated with the indications of some standard instrument such as a counter and stop watch or by use of a chronograph.

(a) **Calibration against a Continuous Counter.** Increasing and decreasing readings of the tachometer should be taken at each speed. The mean of these readings is plotted against the speed as shown by the counter for the calibration curve.
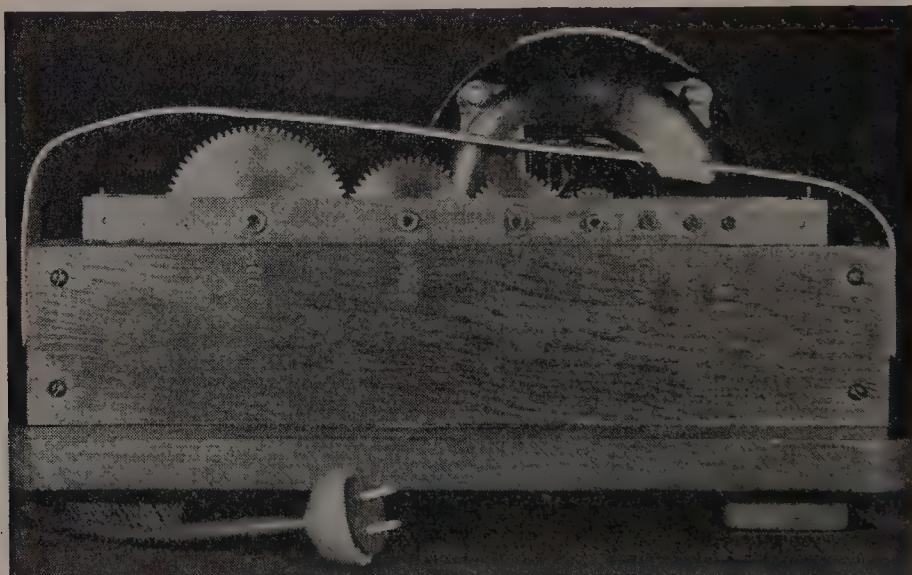
Fig. 33.—Tachometer Tester.

If the continuous counter is arranged on the variable speed shaft so that it can be applied at the same time as the tachometer, the procedure is as follows. Starting with a slightly lower speed than is required, the tachometer is connected and the speed raised to the desired amount. Time and counter readings are then taken with the tachometer still connected. If the speed remains constant as indicated by the tachometer, the observations are valid after the second readings of time and counter are taken. The decreasing readings are obtained similarly.

When the speed is high the counter becomes difficult to read. This may be obviated by an arrangement for throwing the counter in or out of connection with the variable speed shaft.

A hand counter may be used if it can be applied to one end of the shaft while the tachometer is at the other. It is inadvisable to take readings of the two instruments at other than the same time as the speed may vary.

The time during which the revolutions are counted should be of sufficient length to reduce the error of starting and stopping to less than the precision of the tachometer. For instance, if the tachometer cannot be read closer than 5 r.p.m., at any part of its scale, then at 250 r.p.m. the error in reading the instrument is 2 per cent, and the error in reading the time should be made less than this, say 1 per cent. Now a moving number is unlikely to be timed with an ordinary watch closer than ½ sec.; therefore the counting should proceed through at least 50 sec. for the desired precision. Similarly, at 500 r.p.m., the error in reading the instrument is 1 per cent; the starting and stopping error should be reduced to ½ per cent, and the timing should proceed through 100 sec. The use of a stop watch will greatly reduce this time, as it reduces the starting and stopping error to ¹⁄₁₀ sec. or less.

(b) Calibration against a Chronograph. With this apparatus the tachometer may be left in connection with the variable speed shaft, and the speed first increased and then decreased in a series of steps at constant speed. The moving record may be stopped while the speed is being adjusted. The time for counting may be very much reduced, as the chronograph shows the whole number and fraction of turns of the shaft that occur during a time beat which may be as small as desired.

(c) Calibration of a Recording Tachometer. The principles to be observed are exactly the same as given under (*a*) and (*b*). It should be observed that certain types of this instrument have a time lag in their indications behind the true revolutions per minute of shaft whose speed it is intended to measure. That is, when this shaft changes in speed, a certain interval is required before the change is felt at the recorder. This interval should be noted and reported.

TIME

The measurement of time is one of the most important measurements made in the laboratory or in engineering test work, since every test or experiment which involves a rate requires knowledge of the passage of time. The accuracy of time measurements, for engineering purposes, is not usually required to be so refined or precise as in some of the exact sciences, such as physics and astronomy. In certain types of engineering work, however, time measurements of more than ordinary accuracy are required.

Perhaps the most convenient form of timepiece, for laboratory and general work, is the *stop watch*. Considerable care is required in the

handling and use of stop watches to insure their accuracy. They are delicate instruments and should be treated as such; one sharp jar or blow may entirely destroy the accuracy of the watch and some of the delicate parts may be damaged or broken. The better watches have jewelled bearings which are easily cracked by careless handling.

Most stop watches have a movement employing the *lever escapement* similar to that of the usual pocket watch but with the stop and reset devices added. The balance wheel is designed to beat $\frac{1}{5}$ sec. so that, in the better class, it is possible to measure time to the nearest $\frac{1}{10}$ sec. There is a small auxiliary dial which registers the whole minutes.

Fig. 34a.—Ship's Watch (outer case not shown).

There is also on the market an *electric stop clock* which is capable of measuring time to $\frac{1}{10}$ or $\frac{1}{5}$ sec. These are very convenient for use in the laboratory and have the advantage that they can be mounted in a given position and, hence, are not so likely to meet with accidental damage. The accuracy of such clocks depends entirely upon the precision with which the frequency of the alternating current is maintained.

The reader is warned against making adjustments or otherwise tinkering with these timepieces. Usually it is best to have laboratory watches checked periodically by a competent watchmaker or repairman. Furthermore, if the cases are opened frequently, dust and dirt will accumulate in the movements very rapidly. It should be noted that, if a watch of any sort begins to gain, it is usually a sign that the watch needs cleaning. The dirt and gumminess of the oil tend to make the balance wheel "beat short" which increases its period, causing the watch to gain.

Standard Clocks. In order to determine the accuracy of a stop watch, it should be compared with some type of standard clock, which is kept in one place in the laboratory and its rate of gain or loss obtained by comparison with standard time signals. One type of standard clock, suitable for laboratory use, is the *ship's watch* (Fig. 34a). These are oversized watch movements mounted on gimbals in a hardwood case which fits inside of a larger, heavily padded case. The outer case pro-

tects the instrument from vibration and sudden changes in temperature. Resting the outer case on a thick pad of sponge-rubber is an added protection against vibration.

The inner case is provided with a glass cover or window through which the dial may be seen, so that the time may be read without exposing the movement to drafts and sudden changes of temperature.

A more expensive type is the *chronometer,* marine type. It is equipped with a special type of escapement known as the *detent* and is usually designed to beat ½ sec. It is mounted in gimbals in the same way as the ship's watch and has the same inner and outer cases but is generally somewhat larger in size.

Such timepieces are rarely perfect timekeepers in the usual sense of the term; they gain or lose but the rate of gain or loss is, or should be, practically constant and not more than a few seconds per week. To assist in keeping the constant rate, the timepiece should be wound at the same time every day and it should be the assigned duty of some one person to carry out this routine. Some ship's watches and chronometers are equipped with 8-day movements but, even so, they are usually found to run with a better rate if wound daily.

The best type of standard clock for general laboratory use is the weight-driven, escapement clock with a pendulum designed to beat

Courtesy Gaertner Scientific Corp.
Fɪɢ. 34*b.*—Regulator Clock.

seconds (Fig. 34*b*). In the highest grade of these clocks, the pendulum bar is made of *Invar,* an alloy with an extremely small coefficient of

expansion; otherwise, it should have a pendulum which compensates for temperature changes. A mercury cup can be mounted within the case so that, as the pendulum swings by, an electrical contact is made. Signals, so initiated, are used to operate a timing-pen on a chronograph. The best clocks of this type have a variation, when in good adjustment, of less than 10 sec. per month.

There is now available a precision clock which is far more accurate than is necessary in most engineering work. However, where precise time intervals have to be measured, these crystal-controlled synchronous clocks meet the need. They depend for their accuracy upon the oscillations of a piezo-electric crystal which generates weak currents of predetermined frequency. These weak currents are amplified by means of vacuum tubes and are used to drive a synchronous-motor clock. With good control of the input voltage and the temperature of the crystal, such clocks will have an error of only a small fraction of a second per year. The whole equipment is quite massive and fairly expensive.

All clocks, of whatever kind, can be checked by means of *precision time signals* which are broadcast regularly by radio. One of these is the signal sent out four times a day by the Naval Observatory through station NSS at Annapolis and by a number of other stations in the United States and its possessions. Perhaps a more convenient signal, because it is available at any time throughout the 24-hr. day, is that broadcast by the National Bureau of Standards through station WWV. These signals are sent out on frequencies of 2.5, 5, 10 and 15 megacycles and consist of two tones (continuous) of 440 and 4000 cycles per sec. and a pulse once each second except the 59th sec. of each minute.

The tones are interrupted precisely on the hour and each 5 min. thereafter and are resumed precisely 1 min. later. During this period the time is given, Eastern Standard Time, in telegraphic code and, in addition, a voice announcement is made each hour and half-hour explaining the system of signals. Even a small and inexpensive short-wave receiver can be used to pick up these signals.

The *chronograph* is an instrument by which a graphic record of time is made. The instrument is shown diagrammatically in Fig. 35. Paper is rolled over a drum, D, by means of a clockwork or electric motor, and in contact with the paper is a pen on a light arm, A, which vibrates at regular time intervals by a connection, usually electrical, with a standard clock. Another arm, A', is similarly actuated by connection to the device to be timed. If, for instance, the speed of a shaft is to be measured, a contact is made once every revolution so that the arm, A', will vibrate

once for each turn of the shaft. Meanwhile, the pen, A, is marking off seconds and, by counting the number of revolutions indicated in a given number of seconds, the angular velocity of the shaft is determined.

Where it is desired to have the time-line show intervals shorter than 1 sec., an electrically sustained tuning-fork may be used. Tuning forks which vibrate at 100 cycles per sec. are used frequently and are made to actuate the timing pen through appropriate electrical connections. Tuning forks may be had which hold their frequency to a high degree of precision.

FIG. 35.—Chronograph.

In cases where it is desired to time a number of simultaneous or related events, the chronograph may be fitted with a number of pens. The occurrence of an event will actuate one of the pens so that its relation to other events may be seen and measured on the record.

The chronograph is an extremely useful instrument and where accurate timing of small intervals is essential it is indispensable. It also has the advantage of giving an automatic and permanent record.
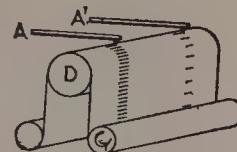
12. TEST OF A STOP WATCH

(a) **Comparison with a Standard Clock.** The observer should hold the watch in his hand and observe the second hand of the standard clock. Stand close enough to the standard clock so that the ticks may be heard clearly. Count the ticks for a few seconds, in order to become accustomed to the rhythm, and, as the second hand crosses a convenient mark on the dial, start the watch. Be careful not to anticipate the instant when the hand crosses the mark. The watch may be stopped in a similar fashion after the desired amount of time has elapsed.

There is a chance of personal error in this method due to personal time lag. However, the chances are that the same amount of lag will be present in stopping the watch as there was in starting. If the two time lags are equal they will compensate.

It is best to test a stop watch over several different periods of time; such as 1 min., 2 min., 5 min., 10 min., and so on. This will make sure that the watch will not stick or slow down. Occasionally dirty or imperfect movements will stop or slow down after the watch has run a few minutes. This is decidedly inconvenient when the watch is used in test work.

Watches should also be tested for position. The usual positions are: face up, face down, pendant left, pendant up, and pendant right. If great variations are found, due to position, the watch should be sent to the maker or a competent repair man for adjustment. The same is true of the adjustment for temperature. If watches are to be used in very warm or very cold places, they should be tested for the effect of temperature
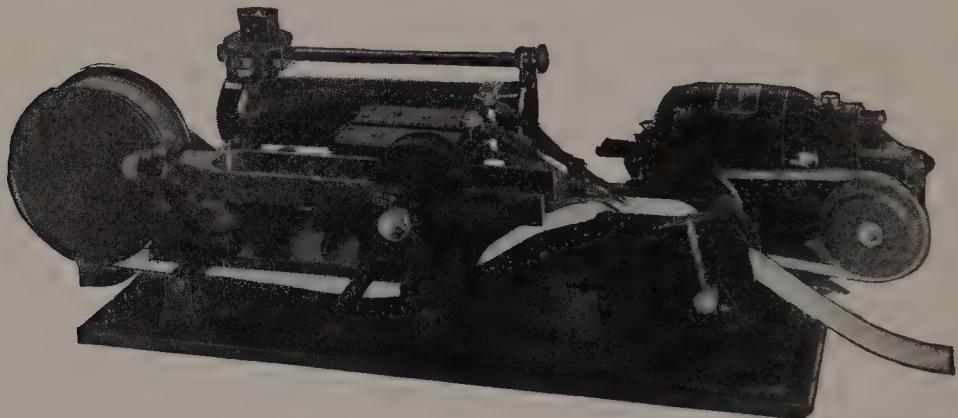
Courtesy Gaertner Scientific Corp.

Fig. 36.—3-Pen A354 Gaertner Chronograph.

and, if great variations are found, they should be adjusted by a competent person.

(b) Testing with a Chronograph. In making the test of a stop watch with a chronograph, the watch should be mounted in a frame and a mechanism arranged which will actuate both the watch and the event-timing pen of the chronograph. One of the other pens is connected to a standard clock.

The procedure is identical with the method described in (*a*). This method gives much greater accuracy and is preferred if the necessary equipment is available.

IRREGULAR AREAS AND MEAN HEIGHTS

Numerous engineering instruments, of which the engine indicator is one, have been devised to give an autographic diagram of the measured quantity expressed as an ordinate, the abscissæ being generally time or in linear units simply. From such a diagram often a mean value of the measured quantity is desired, that is, the mean ordinate of the diagram to scale. For this purpose, *planimeters* are used. Some planimeters measure the mean ordinate directly; others measure the area of the

diagram, from which the mean ordinate may be found by dividing by the length. The best known of these two types are the Amsler Polar planimeter and the Coffin averaging instrument. The former is used to measure any irregular area; the latter is applied chiefly to the determination of the mean height of indicator diagrams.

13. THE POLAR PLANIMETER

Principles. Fig. 37 shows in diagram the Amsler planimeter. When the tracing point traverses any closed curve, 1–2–1, the record wheel follows in a certain path and, through contact with the surface upon which
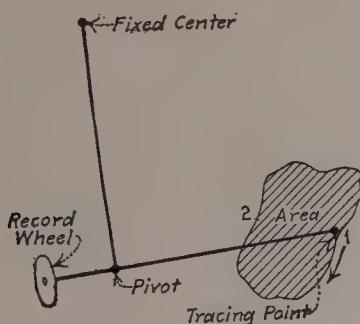
FIG. 37.—Polar Planimeter.   FIG. 38.
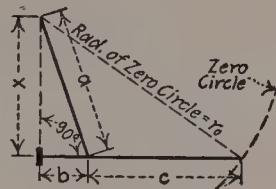
it bears, is given a motion partly rolling and partly sliding. The principle upon which the instrument depends is that the rolling motion of the wheel is directly proportional to the area circumscribed by the tracing point. If, then, the wheel is properly graduated, the area may be read directly.

When the area to be measured is comparatively large, the fixed center of the instrument is placed within the area, thus securing a greater reach of the tracing point (see Fig. 41). In this case, the readings of the wheel must be interpreted differently and the constant known as *the zero circle* must be known. The zero circle may be defined as one of such size that if the fixed center of the planimeter is placed at its center, and if the tracing point then traverses its circumference, there will be no motion of the record wheel. It may be seen from Fig. 38 that this condition requires that the plane of the record wheel shall pass through the fixed center, for in this position the wheel will travel in the direction of its own axis and there can therefore be no rolling.

The algebraic expression for the radius of the zero circle may be deduced as follows (see Fig. 38):

$$r_0{}^2 = x^2 + (b + c)^2$$
$$= (a^2 - b^2) + (b^2 + 2bc + c^2)$$
$$= a^2 + c^2 + 2bc \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

To deduce the mathematical relation between the area circumscribed by the tracing point and the circumferential motion of the record wheel, there are two cases, namely, the fixed center outside and the fixed center inside the area circumscribed. The following paragraph gives an outline of the deduction for the first case.

(1) The motion of the tracing point is referred to polar coordinates whose pole is at the fixed center of the instrument. A polar differential of area is considered as the one circumscribed by the tracing point. An algebraic expression for this area is obtained in terms of the polar coordinates. (2) An expression is deduced for the motion of a point on the record wheel cricumference in terms of the constants of the instrument. The motion is that which takes place when the tracing point has outlined the differential of area. (3) By comparing the two expressions (for the area and for the motion of the wheel) the desired relation is obtained.

Fig. 39 shows the differential of area, marked 4–5–6–7. Using the notation of the figure,

$$\text{Area } 1\text{–}4\text{–}5 \quad = \tfrac{1}{2}r \cdot r dK = \tfrac{1}{2}r^2 dK$$
$$\text{Area } 1\text{–}7\text{–}6 \quad = \qquad\qquad \tfrac{1}{2}r_1{}^2 dK$$

Subtracting, $\quad \text{Area } 4\text{–}5\text{–}6\text{–}7 = \qquad \tfrac{1}{2}dK(r^2 - r_1{}^2) \quad . \quad . \quad . \quad (2)$
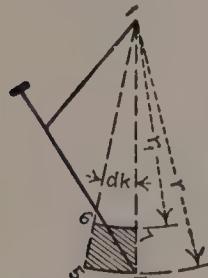
This is the expression for the differential of area.

FIG. 39.

Consider now the corresponding motion of the record wheel. It is to be observed first that this is not affected by the radial motion of the tracing point. The angle between the arms when the point is at 4 (Fig. 39) is the same as that at 5. The same applies to points 6 and 7. Hence, the motion of the wheel when the tracing point passes on the radial from 5 to 6 is the same as that when it passes from 7 to 4. But as these two motions are opposite, they neutralize each other. The same reasoning applies to any irregular area as in Fig. 37. The radial component of the motion from 1 to 2 is the same in amount as that caused in passing from 2 to 1, but opposite in direction. So we may altogether disregard the motion of the wheel produced by the radial motion of the point.

Considering again the differential of area of Fig. 39, it is seen that the motion of the tracing point from 4 to 5 is greater than that from 6 to 7, and that the angle made by the arms is different when the point traverses the two arcs. Hence the circular component of the motion of the tracing point when traversing a closed curve produces a record on the wheel. We have, then, to consider the effect of this component.
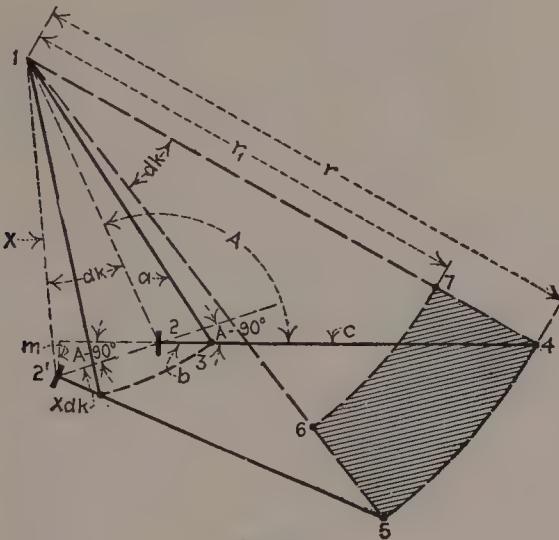
Fig. 40.

Refer to Fig. 40. The record wheel moves from 2 to 2' when the tracing point passes from 4 to 5. The rectangular component of the motion, 2–2', causing rotation of the wheel, is represented by the line m, perpendicular to the wheel axis. m is therefore the distance moved by any point on the record wheel circumference relative to its axis. In the figure it was necessary to draw dK as a finite quantity for the sake of clarity. Actually it is infinitesimal and, therefore, the angles between the base and sides of the triangle, 2–1–2', are sensibly right angles. From this it follows that:

$$m = XdK \sin (A - 90°)$$

$$= -X \cos A dK \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

From 1–2–3, $a^2 = b^2 \qquad + X^2 - 2bX \cos A.$

From 1–2–4, $r^2 = (b + c)^2 + X^2 - 2(b + c)X \cos A.$

Subtracting, $a^2 - r^2 = -2bc - c^2 \quad + 2cX \cos A.$

from which, $X \cos A = \dfrac{1}{2c} (a^2 + c^2 + 2bc - r^2).$

From (1),
$$= \frac{1}{2c}(r_0{}^2 - r^2) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Combining (3) and (4)

$$m = \frac{dK}{2c}(r^2 - r_0{}^2), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

which is the value for the rotation of a point on the circumference of the record wheel when the tracing point moves from 4 to 5. Similarly, when it moves from 6 to 7, the motion is

$$m_1 = \frac{dK}{2c}(r_1{}^2 - r_0{}^2).$$

The difference between these two motions, m and m_1, is the resultant motion M of a point on the circumference of the record wheel when the whole area has been circumscribed, or

$$M = m - m_1 = \frac{dK(r^2 - r_1{}^2)}{2c}. \quad . \quad . \quad . \quad . \quad (6)$$

Comparing (2) and (6)

$$\text{Area } 4\text{--}5\text{--}6\text{--}7 = \frac{dK}{2}(r^2 - r_1{}^2) \cdot \frac{c}{c} = Mc, \quad . \quad . \quad . \quad (7)$$

FIG. 41.

which is the required relation.

The deduction of the relation for the second case, namely, when the fixed center is within the area to be evaluated, is similar. Under this condition, however, the circular component of the tracing point is always in one direction. Hence there is no subtraction as indicated by equation (6) and the differential expression is
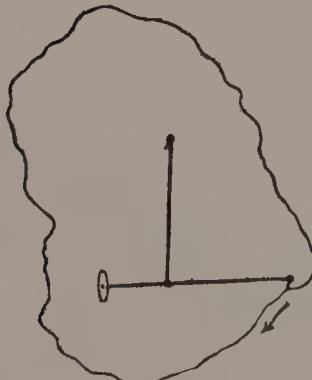
$$M = \frac{dK}{2c}(r^2 - r_0{}^2).$$

Integrating this for the entire circumferential motion of a point on the circumference of the record wheel,

$$M = \int_0^{2\pi} \frac{dK r^2}{2c} - \int_0^{2\pi} \frac{r_0{}^2 dK}{2c} = \int_0^{2\pi} \frac{d(\text{area})}{c} - \int_0^{2\pi} \frac{r_0{}^2 dK}{2c}$$

$$= \frac{\text{area}}{c} - \frac{2\pi r_0{}^2}{2c}.$$

From which, $$\text{Area} = Mc + \pi r_0{}^2, \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (8)$$

which is the required relation.

From these two deductions it is seen that,

First. When the fixed center of the planimeter is outside the area to be integrated the motion of a point in inches, on the circumference of the record wheel, relative to its axis, multiplied by the length of the arm *c*, equals the desired area.

Second. When the fixed center of the planimeter is inside the area to be integrated, this product added to the area of the zero circle equals the desired area. With the working form of the instrument, the multiplication Mc is not necessary, the wheel being graduated in terms of square inches, or other units of area.

If N is the number of turns of the record wheel on its axis, and w the diameter of the record wheel in inches, these relations may be expressed as follows:

First. $$\text{Area} = N \times \pi w \times c. \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (9)$$

Second. $$\text{Area} = N \times \pi w \times c + \pi r_0{}^2. \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (10)$$

In some types of polar planimeter, the arm *c* is made adjustable in length, so that areas drawn to various scales may be read directly.

As an exercise, the student should set the arm *c* at any random length, and then by traversing a known area and noting the number of turns N, show that equation (9) holds true.

(a) **Determination of the Zero Circle.** If the lengths of the arms, *a*, *b*, and *c*, are carefully measured, the radius of the zero circle may be computed by means of the relation given by equation (1), and from this the area.

If the axis of the wheel is in a perpendicular plane through the arm *c*, the graphic method suggested by Fig. 38 may be used. Two lines are drawn at right angles to each other, the fixed center placed on one, the tracing point on the other, and the point of contact of the wheel and the paper at the intersection of the lines. The radius of the zero circle is then the distance between the fixed center and the tracing point. If the axis of the wheel does not lie in a vertical plane through the arm *c* (as is sometimes the case) the construction must be altered to allow for this difference.

When the value r_0 is found, it may be checked by noting a zero motion of the wheel when the tracing point moves as in Fig. 38. The arms may be clamped by placing the fixed and tracing points in two holes

pierced in a strip of Manila paper, these holes being distant from each other an amount equal to the radius of the traversed circle.

(b) Comparison of Instrument Indications with Known Areas. For this purpose may be used a check rule, an instrument by which the tracing point of the planimeter may be swung through a circle of known radius. Initial and final readings of the record wheel are taken; the difference between these readings should equal the area traversed. When the fixed center of the planimeter is inside the area, the difference between the readings should be added to the area of the zero circle.

If the instrument indications do not correspond to the actual areas, the length of the arm, c, should be adjusted.

Notice that the wheel reads positively if the direction of the tracing point is clockwise, *with the single exception of the case when the area to be integrated is less than the zero circle and the fixed center is inside the area,* in which case subtract from area of zero circle.

It is well always to traverse the area in a clockwise direction and to use the following forms:

$$\text{Area} = \text{second reading} - \text{first reading,}$$

or

$$\text{Area} = \text{second reading} - \text{first reading} + \pi r_0^2.$$

As an exercise record the results of two or three area readings of the following and compare with their calculated values.

> Small circle, fixed center outside.
> Circle > zero circle, fixed center inside.
> Circle < zero circle, fixed center inside.

Note carefully. Do not put down the result of an area only, but record in tabular form, as follows:

1st Reading.	2d Reading.	Result.

(c) Comparison of Instrument Indications with Areas to Scale. Planimeters with adjustable arms generally are arranged so as to be applicable to various scales. For instance, for a certain adjustment, each graduation of the wheel may indicate 1 sq. ft. on an area drawn to a linear scale of $\frac{1}{4}$ in. equals 1 ft. For comparison the check rule may be used, its area in square feet on a $\frac{1}{4}$-in. scale being figured by multiplying its area in square inches by the square of the linear scale, that is, 16.

As an exercise, compare the planimeter and calculated results of a circle larger than the zero circle, using the scale setting.

(d) Arm Adjustment to a Required Scale. Suppose it is required to read areas to a scale not provided for by the markings of the adjustable arm. The necessary length of the arm c may be calculated from equation (7). In this equation, we may assume the area to be 1 sq. in. Then the motion M equals the length of one graduation on the wheel multiplied by the number of graduations previously selected to represent the scale units in 1 sq. in.

Let w = diameter of record wheel, inches;
$\quad G$ = number of graduations on the record wheel;
$\quad X$ = number of graduations representing one scale unit of area;
$\quad Y$ = number of scale units of area in one square inch.

Then, length of one graduation $= \dfrac{\pi w}{G}$ in inches.

When the tracing point traverses 1 sq. in. of area, the number of graduations corresponding to the motion, M, will be XY; and

$$M = \frac{\pi w}{G} XY.$$

From (7)

$$\frac{\pi w}{G} XYc = 1 \text{ sq. in.},$$

and

$$c = \frac{G}{\pi w XY},$$

in which everything is known except c, which may be found.

A convenient application of this principle is in the direct determination of horsepower from indicator diagrams, for which purpose the arm c should be adjusted to the length given by the following formula.

$$c = \frac{10500 lG}{SKwX}, \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (11)$$

in which G, w, and X are as previously defined, and

$\quad l$ = length of the indicator diagram, inches;
$\quad S$ = scale of the indicator spring;
$\quad K$ = product of LAN in the $PLAN$ formula.

14. The Coffin Planimeter

Principles. Fig. 42 represents this planimeter. It consists of a single arm one end of which carries the tracing point, the other end bearing on a pin which slides in a guide represented by the line gg. The record wheel is mounted on the arm as shown. An examination of Figs. 42 and 37 will show that the Coffin planimeter is the same in principle as the polar; the mechanical difference being that the arm a of the former is
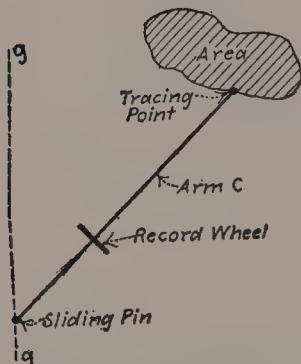
FIG. 42.—Coffin Planimeter.   FIG. 43.
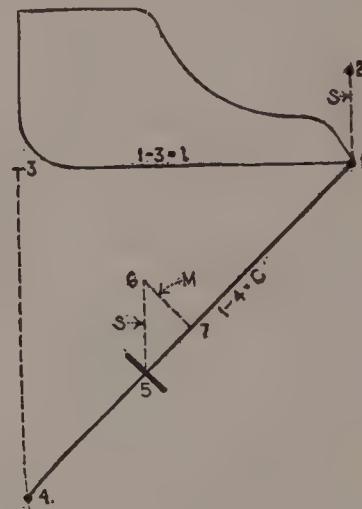
infinitely long, so that the pivot swings in a right line instead of the arc of a circle. The relation Mc = area therefore applies (see Test 13).

The instrument may be used for finding areas, but its chief use is in determining the mean heights of indicator diagrams. For this purpose the indicator diagram is arranged as shown by Fig. 43 with its left-hand extremity in the line of the guide, and its base perpendicular to the line of the guide. The tracing point of the planimeter is then placed at the extreme right-hand point of the diagram, 1, and an initial reading taken. Next, the figure is traced in a clockwise direction, until the tracing point comes back to the point, 1. If now the tracing point is moved on a line parallel to the guide line until the wheel indicates again its initial reading, the distance 1–2 thus traveled by the tracing point equals the mean height of the indicator diagram in inches.

To prove this, it should first be mentioned that the motion of the wheel corresponding to the motion of the point from 1 to 2 is the same

as that corresponding to the motion of the point around the area, since the wheel returns backward to its initial position during the motion 1–2.

In Fig. 43, the motion 1–2 of the point causes a change of position of the wheel from 5 to 6. The line 6–7 is the component of this motion which causes rolling of the wheel. Since this rolling is the same as that taking place when the area is circumscribed, it equals M in the formula Mc = area, the area being that of the indicator diagram, and c the length of the planimeter arm (see Test 13, principles). Also,

$$\text{Area} = h \times l$$

in which h is the mean height and l the length of the diagram. Hence

$$Mc = hl. \text{ from which } h = \frac{Mc}{l}.$$

From the similarity of triangles 1–3–4 and 6–7–5,

$$\frac{S}{M} = \frac{c}{l}, \text{ from which } S = \frac{Mc}{l}.$$

Hence $S = h$, that is the distance 1–2 equals the mean height of the indicator diagram.

(a) **Comparison of Records with Known Mean Heights.** Instead of an indicator diagram, a carefully laid out rectangle may be used. The tracing point should be started in the lower right-hand corner, and the figure traced back to this starting point. The parallel motion necessary to bring the wheel back to its initial reading will take the point back to the upper right-hand corner of the rectangle, if the instrument indicates correctly. This test should be repeated several times, care being taken that the point pursues closely the path of the rectangle.

If the instrument does not indicate correctly, it may be because the length of the arm has been altered, through bending of the tracing point or otherwise, because of faulty graduations of the wheel, or because of the wheel sticking instead of revolving freely.

15. THE AVERAGING OF CIRCULAR CHARTS

Principles. The growing use of autographic instruments of precision yielding circular charts has led to the development of appropriate methods of averaging and integrating. Recording instruments are used both for the measurement of quantities which need not be totaled, such as

pressures, temperatures, CO_2 percentages; and for time rates, such as cubic feet of steam or pounds of water per minute. It is often desirable to average such records, and it is essential, in the case of time-rate ones, to get total quantities. The latter may also be had, when the *average* rate is known, by multiplying this rate by the time. The problem, then, resolves itself into one of finding averages.

It should be noted, however, that there are available special planimeters yielding total quantities from time rate charts, and that many
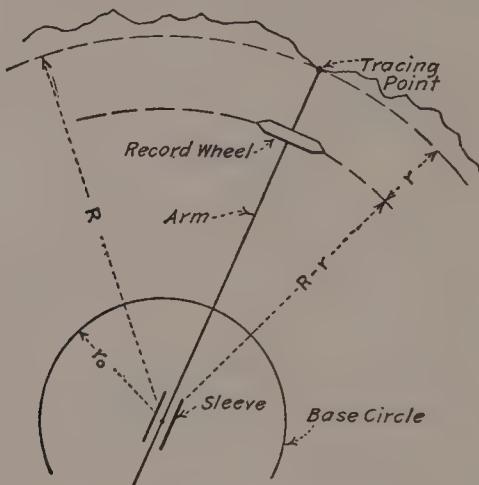
Fig. 44.—Radial Planimeter.

recorders are equipped with automatic ones, so arranged as to be driven by the same mechanism that moves the chart. Averaging methods only will be considered here.

Circular diagrams always have uniform angular coordinates, since they are obtained by clockwork moving in proportion to time. The radial coordinates, on the other hand, may be uniform or nonuniform, depending upon the law controlling the pen movement and its mechanism. When the ordinates are nonuniform, the usual methods of averaging do not apply.

(a) **The Radial Planimeter.** Fig. 44 illustrates the principle of the Bristol-Durand instrument for averaging circular charts. This consists of an arm carrying a record wheel and tracing point at one end, arranged to slide in a pivoted sleeve. With the pivot set at the center of a circular chart, the tracing point can traverse any curve on the chart, the arm sliding in the sleeve when the point moves radially. Since the record wheel axis is set on a radial line, the motion of a point on the wheel circumference with relation to its axis will be due only to circumferential

motion of the tracing point. Radial motion of the tracing point will cause no motion of the wheel on its axis. Thus it is seen that it is only the circumferential component of the motion of the tracing point which causes rolling of the wheel, and since the circumferential component is proportional to the radius, it follows that the rolling will be proportional to the radius.

To deduce the equation of the instrument, let

r = average distance of tracing point to pivot center = average radius
    of curve, inches;
r = distance between tracing point and plane of wheel, inches;
r_0 = distance between center of chart and zero of circular coordinates,
    inches;
d = diameter of wheel, inches;
n = number of turns of wheel when tracing point traverses a given
    curve;
f = fraction of the time included between the curve limits to the time
    represented by a complete revolution of chart;

Then the motion of a point on the wheel circumference will be

$$\pi dn = 2\pi (R - r) \times f,$$

the quantity on the right being the circumferential component of the wheel motion which exclusively produces rolling. Transposing,

$$R - r = \frac{\pi dn}{2\pi f} = \frac{dn}{2f}.$$

If r is made equal to r_0 (adjustment for this is provided), the value of $R - r$ is the average radius of the curve in inches, measured from the base circle. Multiplying by the scale of radial coordinates gives the desired mean.

If the tracing point is *between* the wheel and the pivot, the equation becomes

$$R + r = \frac{dn}{2f}.$$

To determine the mean height from the base circle, in this case, it is necessary to multiply the number of turns, n, by $d/2f$ and subtract from their product r and r_0.

Calibration curves for ready use can be made with charts of given form.

When the radial coordinates are curved (that is, produced by a pen swing around a center), an error will result if the ends of the curve do

not join. To obviate this, the tracing point must be brought back to the same radial distance from the chart center at the finish as that at the start; and this closing must be made by following a curved radial coordinate.

The instrument may be tested for accuracy, first, by tracing a circle of known radius; and, second, by noting whether a wholly radial movement of the tracing point rolls the wheel.

(b) **Approximate Average with Integrating Planimeter.** This instrument is not appropriate to circular diagrams as might at first appear. If the area of such a diagram be found with the polar planimeter and divided by its angular measure expressed in radians, and the square root of this quotient taken, the result will be the square root of the mean of the squares of the radii, instead of the arithmetical mean. However, in cases where the record is not that of a very variable quantity, these two values are not materially different, so that the one can be taken in lieu of the other. The procedure is as follows:

Close the gap at the ends of the curve to be averaged by radial lines, and then find the area bounded by the curve and the radials. Then, if f is the fraction of the time included between the radials to that represented by a complete chart revolution, the mean height of the curve from the center of the chart is (approximately)

$$R = \sqrt{\frac{\text{Area}}{\pi f}}.$$

The average quantity in scale units is now found by noting the value of a division on the chart at a distance from the center equal to this mean height.

POWER

Power is defined as the time rate of doing work. Quantitatively, work is the product of a force and the distance through which it acts, the unit being foot-pounds. A horsepower is defined as 33,000 ft.-lb. of work done in 1 min. If an engine can deliver 33,000 ft.-lb. of work in 1 min., it is rated as 1 hp.; if 66,000, 2 hp.; and so on.
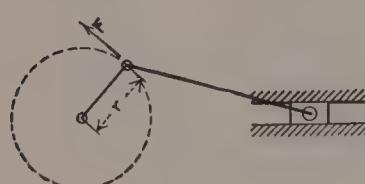
Fig. 45.

Generally engines deliver a rotative effort. Suppose, for example, that an engine transmits an average tangential force of F pounds at its crank pin (see Fig. 45). In one revolution, this force will act through a distance

equal to the circle through which the crank pin has passed, or $2\pi r$, r being the radius of the crank in feet. The foot-pounds of work per revolution are then $2\pi r \times F$, and if there are N revolutions per minute, the work done per minute will be $2\pi rF \times N$, and the horsepower will be

$$\text{hp.} = \frac{2\pi rFN}{33,000}.$$

The quantity rF is called the "torque."

Dynamometers are used for measuring power. Generally the speed is measured independently of the dynamometer by a hand counter or otherwise, so the dynamometer is applied only to measure the torque. This is done by balancing the torque by a measured force acting at a known distance from the center of rotation.

There are two broad classes of dynamometers: *absorption* by which the power to be measured is converted into some form in which it cannot be used; and *transmission,* by which the power is passed on unchanged.

16. Constants of Friction Brakes

Principles. Friction brakes may be used as absorption dynamometers, and of these the **prony brake** is the commonest and most accurate form
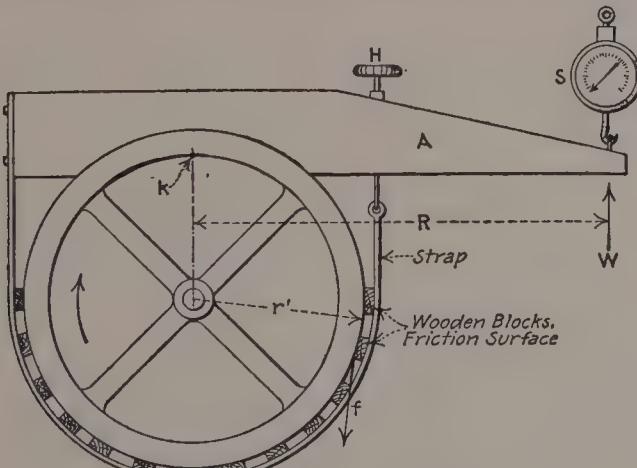
Fig. 46.—Prony Brake.

(see Fig. 46). It consists of a band wrapped around a pulley on the shaft whose power is to be measured, so arranged that by tightening a hand wheel, H, the friction between the wheel and band can be controlled. The band is held from turning by means of an arm, A, attached to it, and

supported by some force-measuring device at its free end. Considering the friction between the band and wheel as a single force, f, then the horsepower developed is $2\pi r'fN \div 33,000$; r' being the radius of the wheel in feet. Now since the force W exerted by the scales, S, produces equilibrium, by the principle of moments

$$RW = r'f,$$

and therefore, by substituting in the expression for horsepower just given,

$$\text{hp.} = \frac{2\pi RWN}{33,000} = BWN.$$

In practice, if the force, W, and the revolutions per minute, N, are measured, the horsepower may be calculated, the length of the arm being known. B is called the "brake constant."

It should be noted that the arm, A, by its weight produces a force acting on the scales which should not be included in the force balancing the frictional effort. This should be allowed for by determining the "unbalanced weight" of the brake, or "brake zero," and subtracting it from the scale readings. Methods of determining the brake zero will be given later.

The Rope Brake is a modified form of prony brake. Fig. 47 shows such a one, the friction between the rope and the wheel being balanced
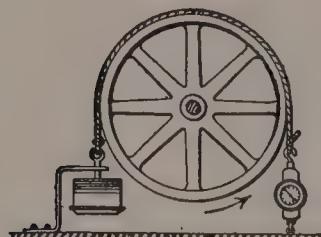
by the force of the scales on the right less the weight on the left. The difference between these quantities is the value of W in the formula, and the value of R in the radius of the wheel plus the radius of the rope. There is no unbalanced weight if the lengths of rope on each side of the pulley are equal. The friction, and therefore the horsepower, may be varied by increasing the weight on the left or by taking more turns of the rope about the wheel; either procedure greatly increases the friction.

FIG. 47.—Rope Brake.

Fig. 48 gives two other arrangements of rope brake, and their equations.

A stout anchor rope, or some form of safety stop, should be provided for both prony and rope brakes. This is to prevent the brake being thrown backward in case the flywheel should reverse during starting and stopping. This frequently happens with gas engines and steam engines having poorly set valves. With large brakes, without safety stops, there is not only the danger of wrecking the brake but also of seriously injuring nearby persons.

In operation, the heat generated by the friction is removed by circulating water within the brake wheel, the latter being provided with internal flanges for that purpose. It is generally sufficient to feed the water at a rate just equal to the loss by steaming.

Hydraulic Friction Brakes are the same as those just described in principle, but quite different in detail. A series of discs mounted on the power shaft revolve in a casing filled with water under pressure, the casing being free to revolve about the shaft but constrained from doing so by an arm which rests on the scale platform. The friction between the discs and the water causes a turning effort upon the casing which is balanced by
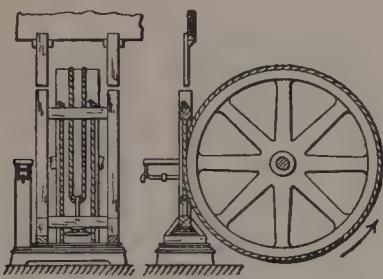
Torque $= (S - \text{Brake Zero})R$ Torque $= (W - S - w)R$
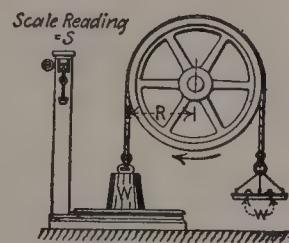
FIG. 48.—Rope Breaks and Their Equations.

a force-measuring device the same as with the common prony brake. Regulation is secured by varying the water pressure. The constants are the same as those for the other types, but in most cases there is no unbalanced weight.

Another type of hydraulic friction brake has a rotor fitted with rows of vanes similar to those of a reaction steam turbine. The casing has rows of plain flat vanes which fit in between the rows of moving vanes on the rotor. When water is admitted to the casing, the moving vanes set the water in motion which reacts on the stationary vanes of the casing tending to cause it to rotate. A radial arm, attached to the casing, with its outer end resting on a platform scales prevents rotation. The computations are the same as in the previous type.

(a) Determination of Unbalanced Weight. If friction could be entirely eliminated between the band and flywheel, the unbalanced weight would be the reading of the scale, Fig. 46. But, no matter how loose the band is, there is always some friction between it and the wheel tending to hold up the brake arm when the wheel is stationary. If the brake is removed from the wheel and supported by a knife edge or circular pin at the point k, Fig. 46, the other end being supported by the scales as in

operation, then the scales will indicate the unbalanced weight provided that the flexible band does not change in its weight distribution. This is one method of finding this quantity.

Another method consists in revolving the brake wheel first in one direction and then in the other, and noting the corresponding readings of the brake scales. With the wheel revolving clockwise, Fig. 21, the scales will indicate the unbalanced weight *plus* the force necessary to balance the friction at the band. Anti-clockwise, it indicates the unbalanced weight *minus* this force, since the friction is reversed. Calling the force balancing friction X,

$$\text{Unbalanced Wt.} + X = \text{1st reading}$$

$$\text{Unbalanced Wt.} - X = \text{2d reading}$$

$$\text{Unbalanced Wt.} \qquad = (\text{1st} + \text{2d reading}) \div 2.$$

When applying this method it is necessary that X remain constant. The band should be quite loose, the bearing blocks wet with oil, and the wheel turned at a uniform rate.

The third method is the same in principle as the second, but the brake is revolved instead of the wheel. The spring balance, Fig. 46, is first drawn upward giving the weight $+X$ reading; then the weight of the brake is allowed to draw it down for the weight $-X$ reading. Note that if the spring balance is slanted out of its correct position relative to the brake, a component force will be indicated.

(b) **Determination of the Horsepower per Pound of Thrust per Revolution.** This is the "brake constant" or $2\pi R \div 33{,}000$. R should be measured with a tape or measuring rod. Note that R is the *perpendicular* distance from the center of the brake-wheel to the line of the balancing force W, Fig. 46.

17. Electric Dynamometers

For testing high-speed machinery the electric generator affords a very convenient form of absorption dynamometer. If the efficiency of the generator is known at any given speed and load, an extremely accurate power determination is possible. If a direct current generator is used the computations are quite simple.

If the stator of the generator is supported so that it is free to rotate, except for a radial arm resting on a platform scale, the power may be determined directly without any of the electrical characteristics of the

generator being known. This is true because the force measured on the scales multiplied by the length of the radial arm gives the true torque, and, if the revolutions per minute be known, the power can be computed readily.

When generators are built, as just described, to be used as dynamometers, the weight of the radial arm is balanced and its length propor-
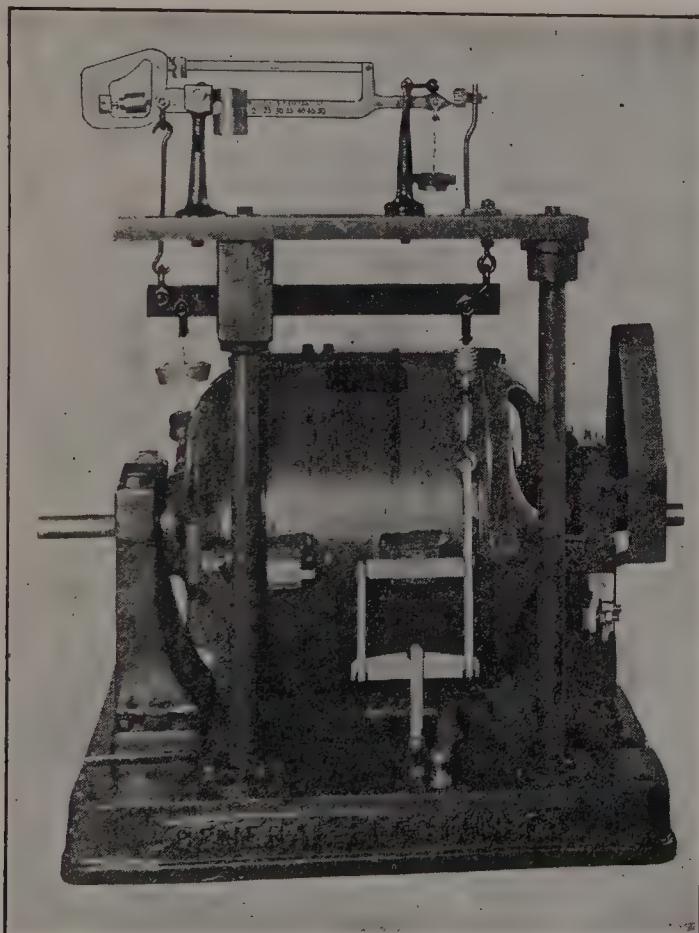
Courtesy General Electric Co.

Fig. 49.—Electrical Dynamometer.

tioned to facilitate computation. The field of the generator is usually separately excited and the output is dissipated in a large variable grid resistance or water rheostat. Variations in load are made by means of the rheostat and by changes in the field strength which changes the output e.m.f. It is best practice to maintain the e.m.f. as high as possible since this keeps the current at a minimum, for a given output, and reduces heating of the rheostat.

An advantage of this type of dynamometer lies in the fact that it may be used to measure the friction horsepower of the machine under test. To do this, the friction and windage losses of the generator must first be measured while it is being run as a motor, which can be accomplished by noting the load on the scales as the dynamometer is run light at different speeds. Then, using the dynamometer to drive the machine under test, a direct measure of the friction horsepower is obtained.

18. CALIBRATION OF A FAN BRAKE

Principles. Fan brakes are convenient for measuring the output of high speed motors that operate at variable speed, such as automobile engines. Fig. 50 shows such a one. The energy of the shaft is absorbed by imparting kinetic energy and heat to the air. With a fixed set of vanes the power thus absorbed varies as the cube of the speed. The capacity of the brake at a given speed of rotation can be changed in only two ways—namely, by changing the size of the vanes or their distance from the center of the shaft. The first method may be taken by swinging the vanes around on their arms so that the effective area resisting motion is lessened. For the second method, the fastenings of the vanes to the arms may permit the desired radial variation.
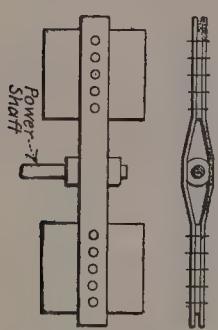
FIG. 50.—Fan Brake.

This type of dynamometer is sometimes mounted on a frame independent of the engine shaft and driven by a belt. In this case, the belt losses and brake journal friction (more or less indeterminate quantities) are added to the resisting effort.

(a) **Calibration against a Transmission Dynamometer.** By means of a transmission dynamometer, simultaneous values of torque necessary to drive the fan and revolutions per minute are determined for a given adjustment of the vanes. From these values, the horsepower absorbed at each speed may be calculated, and a curve of horsepower vs. revolutions per minute plotted. This curve may be used for finding the horsepower, when the brake is in usual operation, from readings of the rotative speed.

(b) **Calibration against a Calibrated Motor.** If the fan brake is belt driven, it may be run by a variable speed electric motor, through the same belt and pulleys to be used when the fan measures power. At a definite speed of the fan, readings of the armature and field currents of

the motor and of its speed are taken. The belt is then removed, a prony brake applied to the motor pulley, and the horsepower of the motor measured under the same conditions of current and speed. If all controllable conditions are the same, this horsepower will be the same as that absorbed by the fan at the applied speed. A series of similar trials at different speeds will give data for a calibration curve.

This calibration will be somewhat in error owing to the fact that the reaction on the motor bearing caused by the prony brake thrust is somewhat different from the reaction due to the belt pull; hence the friction of the motor under the two loads will be slightly different.

(c) Use of the Horsepower Constant. The power absorbed by the fan for a given setting of the vanes equals

$$K \times \text{density of the air} \times \text{(r.p.m.)}^3$$

in which K is a quantity very nearly constant. Assuming that the density of the air also is nearly constant, the product of K and the density may be found experimentally at a convenient speed. Then the horsepower at any other speed may be found, approximately, by multiplying this product by the cube of the speed.

The experimental determination of K times air density is made by taking readings according to either method (a) or (b). Then the desired value equals the horsepower divided by the cube of the applied speed.

Variations in belt losses and in the value of K make this method inexact.

19. Calibration of a Transmission Dynamometer, Weight-Arm Type

Principles. The weight-arm type of transmission dynamometer consists of some device by which the torque of a revolving shaft can be balanced by a standard weight or weights acting with a known leverage. The balancing torque caused by the weights is thus a measure of the horsepower when the revolutions per minute are known. There are many forms of this type of dynamometer differing mainly in mechanical details.

Belt dynamometers of different kinds are in the weight-arm class, of which Fig. 51 represents one. The torque of the transmission shaft is equal to the effective belt pull, $T_1 - T_2$, multiplied by the radius at which it acts, r. Disregarding the friction at the bearings of the pulleys, p and p', the reactions of these bearings will be $2T_1$ and $2T_2$, respectively, as will be seen from a consideration of the belt forces. Taking moments about the fulcrum f, and disregarding the weight of the arm A, we have

$$WR + 2T_2a = 2T_1a$$

from which

$$T_1 - T_2 = WR \div 2a$$

and the torque equals

$$r(T_1 - T_2) = rWR \div 2a.$$
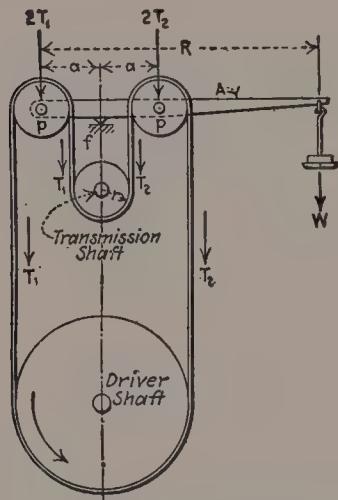
FIG. 51.—Belt Dynamometer. FIG. 52.—Pillow Block Dynamometer.
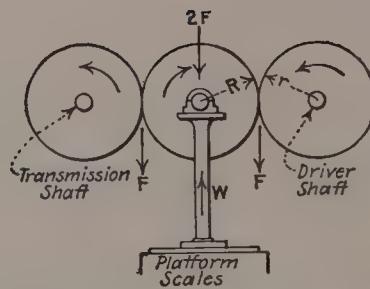
Thus by balancing the belt torque by a known weight W, the power may be measured if R/a is known and if a suitable allowance is made for friction.

A pillow block dynamometer is shown by Fig. 52, consisting of three gear wheels, the middle one of which is mounted on a pillow block so as to bear freely on a weighing device. The turning force of the driver, F, produces an equal resisting force at the gear on the transmission shaft, barring friction. The reaction W, which can be weighed by the scales, thus equals $2F$, and hence the torque of the transmission shaft is

$$Fr = \frac{W}{2} \times r.$$

The reaction W may also be measured by hanging the middle wheel from one arm of a lever above the other carrying a balancing weight.

Some forms of pillow block dynamometer omit the middle wheel and weigh the reaction of the bearing on the transmission shaft directly. When this is done, the transmission shaft must be arranged to rest freely on the weighing device.

The Webber differential dynamometer is a variation of Fig. 52, but similar in principle. It is shown diagrammatically by Fig. 53, which represents three bevel gears, the middle one being supported by a lever whose fulcrum lies on the common axis of the driver and transmission shafts. Similar to the principle shown by Fig. 52 the net force on the middle gear is $2F$. This is balanced by the dead weight W acting with a lever arm of R feet. Taking moments of the forces acting on the lever around its fulcrum we have

$$2Fr = WR$$

from which, torque transmitted =

$$Fr = \frac{W}{2}R.$$
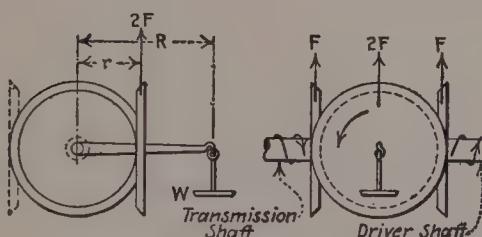
FIG. 53.—Webber Dynamometer.

As actually constructed there is another gear on the lever-arm, as indicated by the dotted line. This does not change the relation, merely substituting for the moment $2Fr$ two moments, $Fr + Fr$. Also there is a jockey weight on the arm in addition to the pan weights similar to the arrangement of a platform scales.

The Emerson power scale is a dynamometer by which the torque is passed to a transmission pulley on the driver shaft through a lever system so arranged that a thrust in an axial direction is given to a nonrevolving sleeve on the shaft. This thrust is balanced by dead weights which consequently measure the torque.

Friction of the moving parts of the dynamometer itself has not been accounted for in the equations. Generally the dynamometer indicates the torque necessary to overcome its own friction and windage in addition to the external torque on the transmission pulley which alone it is the purpose to measure.

(a) **The constants of a dynamometer of the weight-arm type** are, first, the true values of the dynamometer weights; second, the unbalanced weight of the arm or lever system; and third, the leverage ratio produced by the arm or lever system.

The true values of the weights should be determined with a scales sufficiently precise to keep within a reasonable percentage of error.

The unbalanced weight of the arm may be found, as for a prony brake, by revolving the driver shaft by hand in first one direction and then the other, and noting the resulting force at the end of the arm where the weights are to be applied. A spring balance may be used for this purpose. Half the sum of the two readings equals the unbalanced weight.

The ratio of levers may be obtained generally by direct measurement. In the case of the Webber dynamometer, for instance, only the length of

Fig. 54.—Belt Dynamometer.

R, Fig. 53, is necessary. For the arrangement of Fig. 51, the ratio is R/a, as shown by the equation.

(b) Calibration by Calculation. The torque equivalent to each dynamometer weight acting at the previously determined leverage is found by multiplying its value in pounds by the leverage ratio as shown in the equation for the dynamometer in question. A series of such determinations furnishes data by which the horsepower corresponding to any weight may be found when the dynamometer is in usual operation.

The torque equivalent to the unbalanced weight of the arm should be added to that indicated by the weights applied if the turning effort of the driver on the arm is upward. If it is downward, the power being then gaged by scales instead of weights simply, the torque equivalent to the unbalanced weight of the arm should be subtracted.

(c) Allowance for Friction, Windage, and Centrifugal Force. To allow for friction, a crude but convenient approximation is based upon the assumption that the frictional resistance is the same under all conditions of external torque. If, then, the dynamometer is run with the transmission shaft entirely free the weight to balance the arm gives the correction to be subtracted from the readings in usual operation. Windage is included in this, but, if the dynamometer is to be used at various speeds, similar corrections should be determined at these speeds to allow for the varying value of the windage.

With some dynamometers having revolving levers, bell-cranks, and the like, such as the Emerson power scale, centrifugal force acting on these parts may cause a distortion of the indications. This is allowed for under the windage corrections.
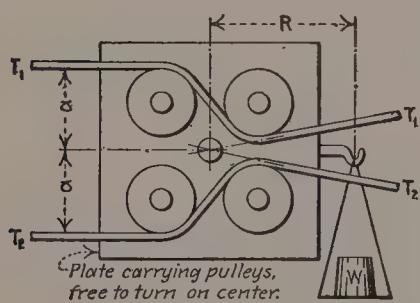
(d) Comparison with prony brake measurements is the most valid means of calibrating any transmission dynamometer. To do this, a prony brake is fitted to the transmission pulley and is adjusted so as to balance each dynamometer weight in turn, power being delivered as in usual operation. The torque shown by the prony brake is the true torque measured by the corresponding weight. When a jockey weight is used as for the Webber dynamometer, a calibration curve or torque against jockey weight positions is convenient.·

If the dynamometer is to be used at various speeds, it should be calibrated at a number of them, the results to be in terms of either horsepower or torque, as shown by the brake, corresponding to each dynamometer weight. In this way friction, windage, and centrifugal force may be taken into account.

20. CALIBRATION OF A TRANSMISSION DYNAMOMETER, SPRING TYPE

Principles. Spring dynamometers differ from the weight-arm type in that the torque to be measured is balanced by a spring or springs through which the torque is passed. The spring is consequently deformed by either a tensile, compressive, or twisting stress. If the constant of the spring is known (that is, the number of pound-feet of torque necessary to cause a unit deformation), then by noting the deformation, the horsepower may be determined.

Fig. 55 shows, in part, the principle of the **Van Winkle dynamometer.** Power is taken off at the loose pulley P which is driven through springs by the disc attached to the driver shaft. The resulting deformation of the springs permits a change of position of the pulley relative to the disc, and this operates a bell-crank lever (not shown) which in turn actuates a pointer on a stationary dial. The dial is arranged to indicate horsepower direct.

Another spring **dynamometer, made by the Central Laboratory Supply Company,** is shown by Fig. 56. Two shafts are connected by a spring through which the power to be measured is passed. These shafts are provided with discs arranged as commutators, being insulated from the shafts except at the shaded portions shown under the brushes. In this position, an electric circuit is completed causing a click in a telephone receiver. The right-hand brush is stationary, but the one on the left is arranged so that it may be swung around the shaft. In operation, the latter is manipulated until a click is heard in the receiver, indicating that both contact pieces are passing under the brushes at the same time. Then

the angular motion of the left-hand brush, shown on the dial and meas-
ured from its clicking position when there is no torque delivered, is equal
to the twist of the spring, and hence is a measure of the torque.
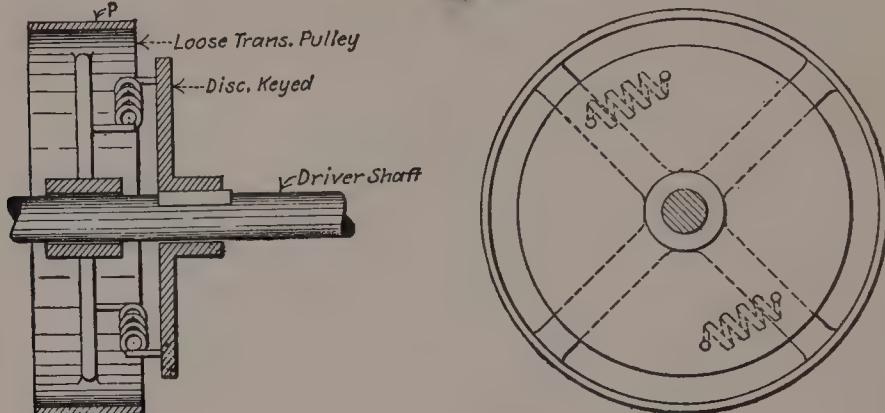
FIG. 55.—Van Winkle Dynamometer.

In usual operation small variations of the torque, due to belt flapping,
etc., make the clicking position somewhat variable. It is therefore con-
venient to read the maximum and minimum angles at which ·no click is
heard and to take the average of these as the twist of the spring.

The Amsler torsion dynamometer is of similar construction except
that a calibrated torsion bar is substituted for the spring. The amount
of twist imparted to the bar is measured by stroboscopic means.
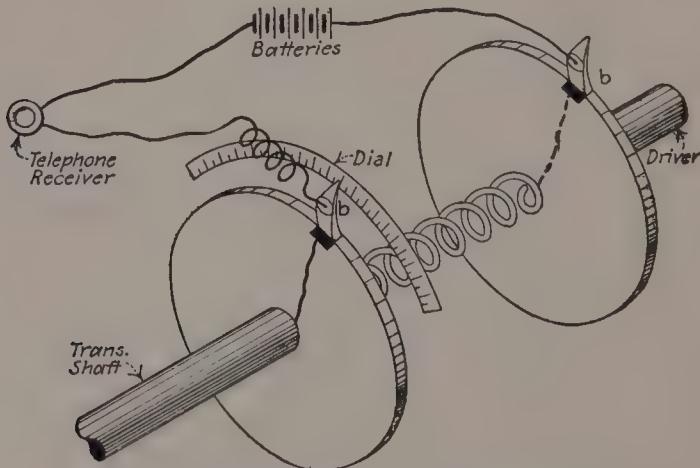
FIG. 56.—Central Laboratory Dynamometer.

The Flather dynamometer (Fig. 57) differs from the foregoing in that
the torque is communicated to the transmission pulley through small

pistons, p, working in cylinders filled with oil. The fluid pressure thus produced is transmitted through tubes, t, and a longitudinal hole through the center of the driver shaft. Connecting with this hole at the end of the shaft is an indicator by which a graphic record of the pressure is made. Since this pressure is proportional to the torque (being produced by the driving force at the cylinders, c, acting at a constant distance from the shaft center) it is a measure of the horsepower.

Shaft Dynamometers. The essential parts of a shaft dynamometer are (a) a calibrated length of shaft or torsion rod and (b) a means of meas-
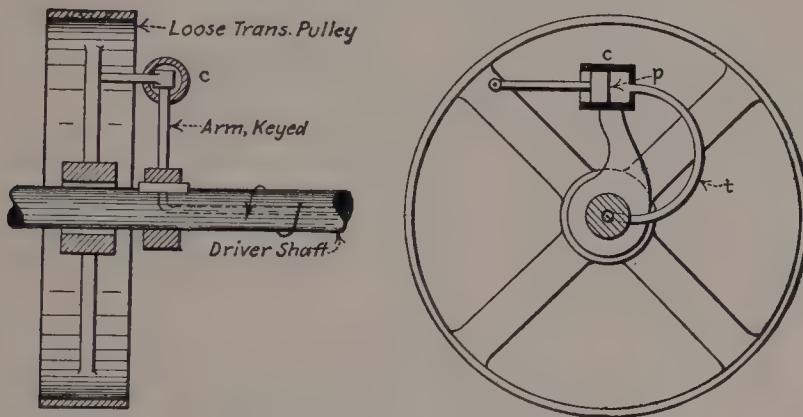
Fig. 57.—Flather Dynamometer.

uring the angular displacement of one end of the calibrated shaft with respect to the other when the shaft is placed under load. The methods of accomplishing (b) are too numerous to be described in detail here. In all types, the calibration results in a dynamometer constant which, when multiplied by the product of the angular displacement and the revolutions per minute, gives the horsepower being transmitted.

Friction affects the indications of a spring dynamometer in two ways. First, since the friction of the loose transmission pulley or of the transmission shaft on its bearings always acts *against* the motion, it makes the indicated torque *greater* than the true external torque. Second, friction of the indicating mechanism makes the indications for increasing torques low, and for decreasing high, similar to the action of friction in a pressure gage. Generally the first effect is greater than the second.

(a) Static Calibration. With the driver shaft clamped, the transmitting shaft may be twisted with a series of known weights acting at a measured distance from the center of the shaft. Thus a series of values

of torque may be obtained with the corresponding instrument indications, the latter being either in dial graduations, degrees of twist, or height of an autographic diagram, as may be appropriate to the instrument tested. To apply the static torque, the transmission pulley may be used as the arm, a rope being tied to a spoke and passed over the pulley face from which to hang the weights. If the necessary number or size of standard weights are not available, a lever may be clamped to the transmission shaft, and a single weight applied at various distances from the center. In this case, the moment of the lever should be accounted for.

Increasing and decreasing values of the torque should be applied and corresponding readings taken to eliminate the effect of friction. For the increasing readings, the weights are caused to bring up the torque gradually to the desired amount. For decreasing, extra torque is brought upon the shaft by bearing on the weights by hand; then gradually removing the hand pressure so that the torque will decrease to the desired value. The average of each pair of readings is then plotted against torque for a calibration curve. By this procedure, for increasing values, the motion of the transmission pulley or shaft is opposite to that for decreasing values. Hence both effects of friction, previously noted, are eliminated.

For dynamometers with which the angle of torsion is read, the curve of torque vs. degrees may be used to determine the spring constant. Then, if S is this constant in pound-feet per degree of twist and A the torsion angle noted in usual operation,

$$\text{hp.} = .00019 \times S \times A \times \text{r.p.m.}$$

which is the horsepower delivered by the dynamometer pulley.

(b) Allowance for Friction, Windage, and Centrifugal Force. The readings of the dynamometer should be reduced by an amount corresponding to the torque necessary to overcome these forces. The values of the corrections may be determined as for weight-arm dynamometers. For recording instruments, a line should first be made on the chart with the dynamometer running free. The diagram under load should then be measured from this friction line as a datum.

For dynamometers using a measurement of the torsion angle, the correction is made in degrees, being subtracted from the reading observed in operation.

(c) Comparison with a prony brake may be made in exactly the same way as for weight-arm dynamometers. (Test 19(d).)

THE ENGINE INDICATOR—REDUCING MOTIONS

The engine indicator is an instrument which makes a graphic diagram giving the relation between the pressure and volume of the fluid in an engine cylinder under working conditions. Since the area of such a diagram is proportional to work, the indicator is a dynamometer of a special type. It is shown in principle by Fig. 58. *C* is a small cylinder with a close-fitting piston *P* which is subject to the same fluid pressure as in the engine cylinder, being connected to it by a short pipe. This pressure compresses the spring *S* until the force of the spring balances the pressure. Thus the motion of the piston is proportional to the pressure. The piston motion is communicated and magnified by means of a linkage *L* bearing a pencil point at *A*. *D* is a metal drum free to oscillate on a spindle and carrying the paper on which the record is to be made. The drum is actuated by the engine cross-head through a cord *K*, a spring within the drum serving to bring it back upon the return stroke. Thus the

Fig. 58.—Engine Indicator.

motion of the drum is proportional to the engine piston and therefore to the volume in the engine cylinder behind the piston. The diagram made by the pencil point on the record paper is one of pressure shown vertically and volume (or piston stroke) horizontally.
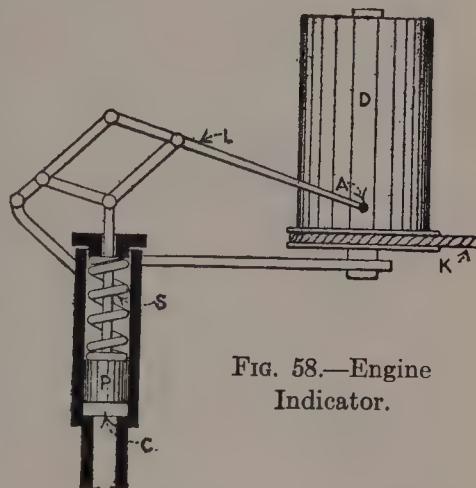
The principal use to which the indicator is put is the finding of the *mean effective pressure* in an engine cylinder throughout its working stroke, from which quantity the *cylinder or indicated horsepower* may be calculated. Fig. 60 is a typical indicator diagram from a steam engine. The average pressure on the forward stroke is the mean height, to scale, of the curve *abc*. On the return stroke, the average pressure (*back* pressure) is the mean height of *cde*. The effective pressure is the difference between these two, or the mean height of the indicator diagram. This may be found by dividing its area in square inches by its length in inches, and multiplying the quotient by the scale of pressure. The scale of pressure is the number of pounds per square inch on the indicator
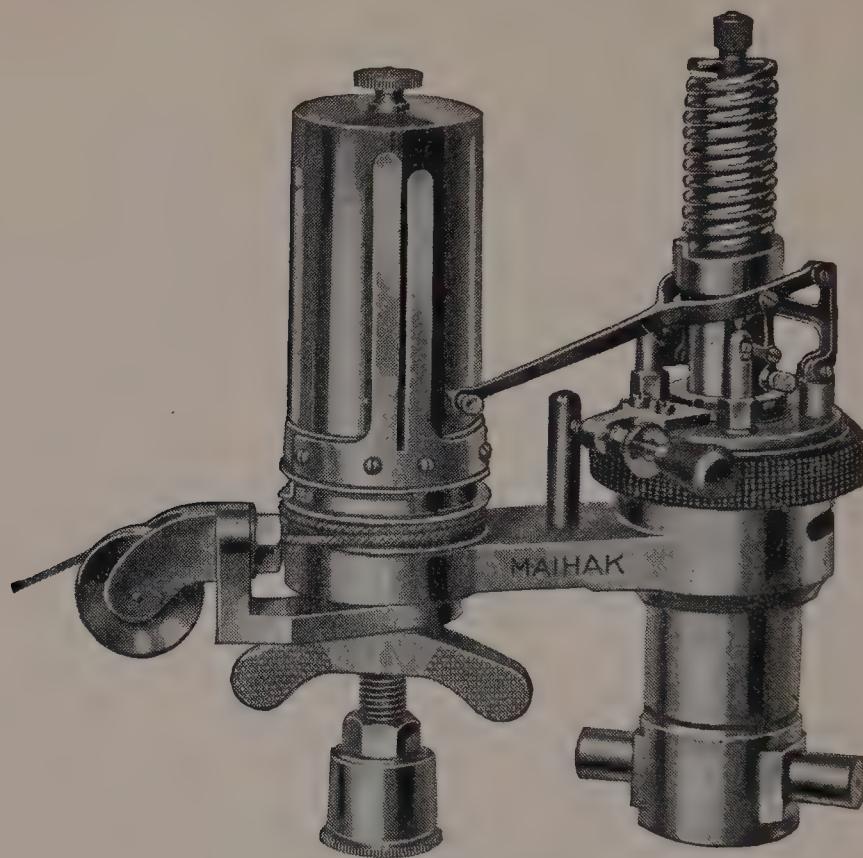
FIG. 59.—Bachrach Indicator.
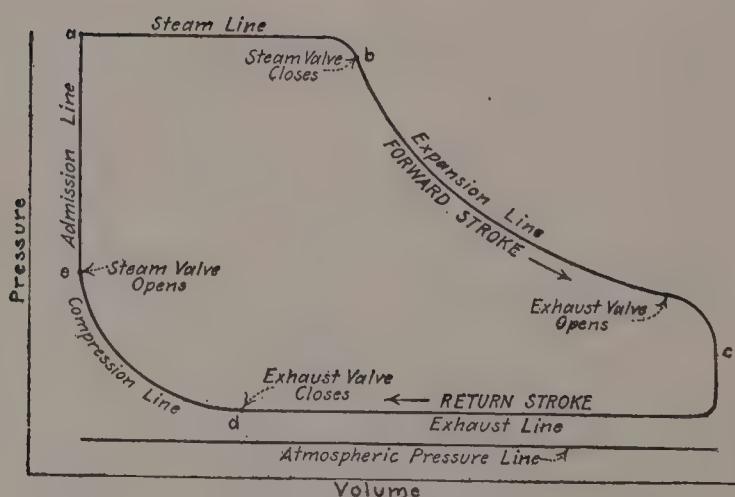
FIG. 60.—Indicator Diagram from Steam Engine.

piston necessary to produce 1 in. of rise of the pencil, and is referred to as the *spring scale*.

The indicator is equipped with a number of springs of different stiffnesses so that one appropriate to given conditions may be selected.

The drum spring is adjustable so that a greater tension may be operative at higher rotative speeds to provide proper acceleration of the drum upon the return stroke.

The cord actuating the drum does not receive its motion directly from the engine cross-head, but from a *reducing motion*, the function of which is to reproduce the engine stroke on a small, but always proportionate scale. A pantagraph is often used for this purpose. Under Tests 23 and 24 are described various types.

21. CALIBRATION OF THE INDICATOR SPRING AND PENCIL MOTION

Principles. There are four causes of error in the ordinates of an indicator diagram. First, when applied to high-speed engines, the inertia of the indicator piston and attached linkage causes a variation from correct positions. Second, at high speed, the pressure in the indicator cylinder lags behind that operating on the engine piston, because of the inability of the steam immediately to traverse the passages to the indicator. Third, the mechanism actuating the pencil may incorrectly magnify the motion of the indicator piston. Fourth, the spring scale may not be exactly known. Friction of the indicator piston and linkage causes a variation of the spring scale, since because of it the pencil is too low when rising and too high when falling.

The first of these errors may be avoided, or reduced, by the use of stiffer springs than are appropriate to low rotative speeds. With a stiffer spring, the total rise of the pencil is less, and therefore the velocity and inertia of the pencil motion parts are decreased.

The second cause of error, lag in the fluid pressure, may be reduced by the use of short and direct pipe connections between the indicator and engine cylinders.

Errors due to the third and fourth causes may be corrected as will be described.

(a) **The pencil motion may be tested as follows:** With the indicator spring removed, a horizontal line is drawn on a piece of record paper placed on the drum, by revolving it by hand, the pencil bearing against the paper at a low position of the linkage. Then, with the drum held stationary, a vertical line is made by moving the pencil and linkage up by

hand. This is repeated with the drum in a second position. The two vertical lines should be parallel and straight, and perpendicular to the horizontal line, if the pencil motion is true.

(b) **Determination of Ascending, Descending, and Combined Spring Scales by Graphic Method.** It is necessary to determine a series of values of true pressures and corresponding heights of indicator pencil. The apparatus for varying and measuring the pressure should preferably be
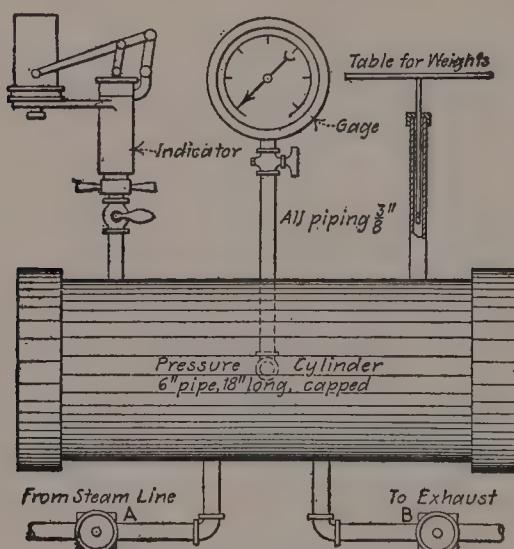
FIG. 61.—Indicator Spring
Testing Apparatus.
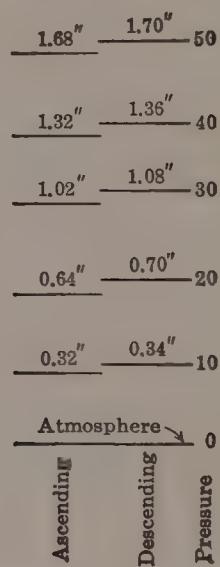
FIG. 62.—Calibration
Records.

one using the same working fluid to which the indicator is subjected in practice, so as to duplicate the conditions of temperature and friction. Dead weights applied directly to the indicator piston are sometimes used, but these do not correctly reproduce the working conditions. Fig. 61 represents a calibration apparatus using steam. The pressure is varied by manipulating valves A and B, an opening of A and closing of B having the effect of increasing the pressure in the large steam chamber, and vice versa. The measuring device is a set of known weights acting on a plunger of known area, from which the pressure balancing the weights may be figured. An accurate and precise Bourdon gage would serve the purpose as well. The gage shown in Fig. 61 is used to indicate the pressure in the chamber when the weights are not balanced, for convenience in manipulating the valves A and B.

Using the calibration apparatus, a diagram similar to Fig. 34b is made on an indicator card. For the ascending pressures, the weight table must

be balanced and rising very slowly when the line is drawn. Similarly, for descending pressures, the table should be gently falling. The table is revolved by hand to reduce friction at the plunger.

The heights of the lines from the atmosphere line are then measured to $\frac{1}{100}$ in., and recorded on the diagram with corresponding pressures as in Fig. 62. These data are then plotted and results obtained as for Test 2a.

(c) Spring Scales by Method of Least Squares. The same experimental data are used, but the results are calculated as for Test 2(b), the value of F in the equations then being the observed pressure in pounds per square inch, and E, the height of the indicator pencil in inches.

(d) Correction of Indicator Diagrams. Various methods, more or less accurate and laborious, have been proposed for applying calibration results to indicator diagrams. Generally, it is sufficient to use the combined spring scale with which to multiply the mean height to determine the mean effective pressure, the error involved being within the limit of accuracy of power tests. But the combined spring scale represents the true scale of the spring more nearly than it does actual conditions, since the method of figuring it eliminates friction. Strictly speaking, the ascending and descending scales should be used separately on the diagram, the former applied to the mean height of *cde,* Fig. 60, since the pencil is rising on that line; and the latter to the mean height of *abc* since there the pressure falls. But when the back pressure line is horizontal in large part, as it almost always is, the descending scale applied to the whole diagram will yield a fair result.

The indicator is sometimes applied to other than power measurements, for which a high degree of accuracy is desirable. It is then necessary to reconstruct the diagram for correct results. A method of doing this is as follows. Calibration data are obtained as previously described, and plotted as shown by Fig. 63. The range of pressure P for the calibration must be the same as that in the indicator diagrams to be corrected, and all operating conditions of the indicator during its test the same as when the diagrams are taken. It will be noticed that the descending curve of Fig. 63 is not straight, and is joined to the ascending curve. Careful experimentation will reveal these characteristics. Actually, there can be no unfilled gap between the two curves, as there would be under the assumption that both are straight lines, an assumption, as was pointed out under Test 2, made only for convenience in approximately figuring the scales.

Having plotted the curves according to Fig. 63, the *ascending* scale

should then be obtained by either of the two methods (a) or (b) previously given. The calibration curve is next laid alongside of the indicator diagram to be corrected as shown by Figs. 63 and 64. Now, the back pressure line is shown correctly to the ascending scale determined from the calibration. To present a point a of the indicator diagram on this scale, the construction abb_1a_1, is used, the point a_1 being the required corrected position of a. Enough points are corrected in this way to re-
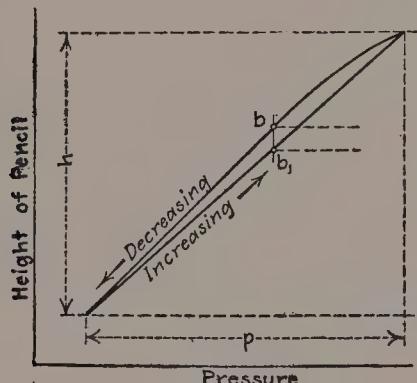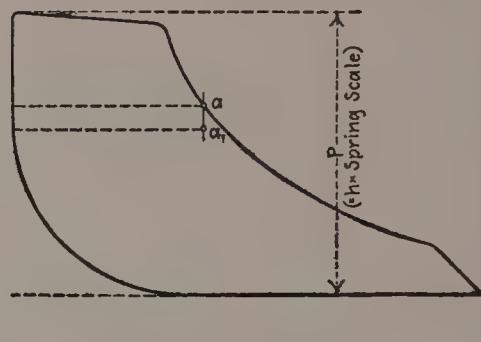
FIG. 63.   FIG. 64.

Correction of Indicator Diagrams.

construct the pressure line of the indicator diagram. The ascending scale will then apply correctly to the whole diagram.

(e) Sampling. When there are a large number of diagrams for a single engine test, a few representative ones only are selected for reconstruction. Suppose, for instance, that there are 24 diagrams and that the average of all their mean heights is H inches. Three of these should be selected as near as possible to this average mean height, and fairly representative in general proportions. These three diagrams are reconstructed and then the desired results obtained from them.

To determine the average of the mean effective pressure from all the diagrams without reconstructing all of them, the value H may be reduced in the proportion that the mean height of the three sample diagrams is reduced after reconstruction. The corrected value of H when multiplied by the ascending spring scale gives the corrected mean effective pressure.

22. Testing the Motion of the Indicator Drum

Principles. When an indicator drum is driven by a cord attached to a reducing motion mechanically correct, the motion imparted to it is

approximately simple harmonic. Upon the forward stroke of the engine, the cord is pulling the drum, and upon the return stroke, the drum spring is stressing the cord to keep it taut. Thus the cord is always under stress. If the stress varies, the cord will stretch according to such variation, and consequently the drum will not assume the correct positions it would have if driven by an inelastic connector.

Stress in an indicator cord is the resultant of three distinct forces: namely, the drum spring tension, the force required to overcome the inertia of the drum, and the force to overcome friction at the drum spindle and at any guide pulleys used to guide the cord. The force of the spring increases with the forward stroke of the drum (the spring being wound up) and decreases upon the return. The variation is usually uniform with the motion.

In general, at the beginning of the forward stroke, inertia increases the cord stress since, as the speed is increasing from zero to a maximum at mid-stroke, inertia effects a tendency of the drum to lag. Beyond mid-stroke, however, the speed is decreasing, and as the drum tends to exceed the velocity induced by the cord, a slackening results. Upon the return stroke the force of acceleration varies in the same way as on the advance.

The force of friction is always opposite to the drum motion and therefore changes its direction at each stroke. During the forward stroke it tends to increase the cord stress, and upon the return, to decrease it.

These three forces are represented graphically by Fig. 65. The resultant stress in the cord equals their algebraic sum at any drum position. At low speeds, the force of acceleration is practically negligible; hence the resultant cord stress increases on the forward stroke, and decreases on the return with lowered values due to the reversal of friction. At high speeds, the force of acceleration reverses this variation as shown by Fig. 65. At some intermediate speed the force of acceleration and the spring tension nearly balance, so that the cord stress is approximately constant during each stroke, but a trifle less on the return because of friction. This is the speed at which the indicator drum is best adapted to work, since the more nearly constant the cord stress is, the less is its stretch and the consequent error.

Let us now compare the effects of a higher speed than this with the ideal condition of constant cord stress. The effect of speed is to increase the stress at the head end and decrease it at the crank end (see Fig. 65). At the head end, therefore, the cord is longer and the drum travels further in this direction. Similarly, at the crank end, the cord is shorter and

the drum is pulled further toward that end. The net effect is to pull out
the indicator diagram. Now, if the diagram were lengthened *uniformly*
its proportions would remain correct, and there would be no error.
Accordingly, if the cord were subjected to a uniformly decreasing and
increasing stress, and if its stretch varied directly with the stress, a cor-
rect diagram would result. It follows that, lacking such uniformity, the
error of any point of the diagram should not be judged by the absolute
stress or stretch at that point, but by the difference between this stress
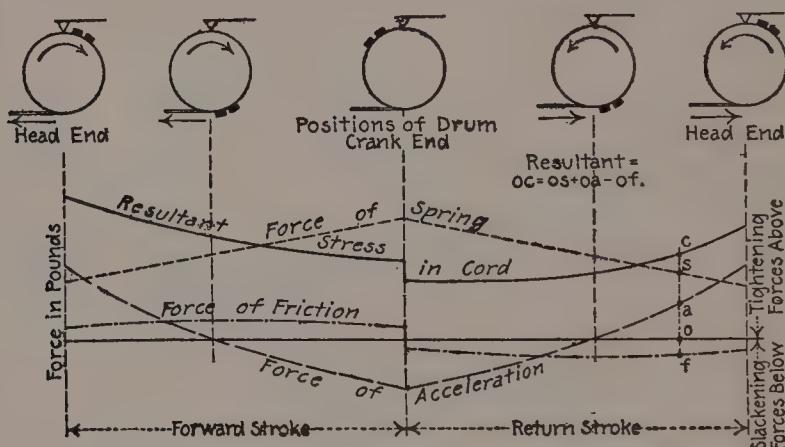
FIG. 65.—Variation of Forces on Indicator Cord.

or stretch and that necessary to produce a uniform lengthening. It is
seen from Fig. 65 that the resultant cord stress is greater than that neces-
sary for uniformity on the forward stroke and less on the return. Hence,
during the forward stroke, the cord is too long and a point on the indicator
diagram is to the left and behind its correct position. During the return
stroke the cord is too short; a point on the indicator diagram is to the
right and again behind its correct position, since the motion is reversed.
The net effect of the stretch of the cord, then, is to make the mean
effective pressure appear smaller, and the cutoff, compression and release
earlier than their true values.

(a) **Testing Indicator Cord.** Tie one end of about 4 ft. of the cord to
be tested to a fixed point on a bench or table, and the other end to a
spring balance. Mark this end a few inches from the balance with a
fine ink line, and under this line place a piece of paper or a foot-rule.
Stretch the cord by pulling the balance horizontally until about 5 lb. are
indicated. Now reduce the force to about 1 lb. and repeat this procedure
a few times. The elongation may then be noted for the applied range

of stress. As the cord in the operation of the indicator is generally not stressed more than 5 lb. or less than 1 lb., this range is appropriate. A good cord should not stretch more than 0.01 in. per ft. per lb., but grades will be found with four times this stretch and more.

The elongation and contraction of the cord are very much greater at stresses less than 1 lb. On this account there may be marked overtravel at the crank end of the drum motion without a visible slackening or vibrating of the cord when the indicator is in usual operation.

(b) **Testing Drum Motion with the Drum Motion Tester.** The apparatus shown by Fig. 66 was devised by the author for this purpose. A shaft S, the

Fig. 66.—Smallwood's Drum Motion Tester.

rotary speed of which may be controlled by a variable speed motor, actuates two similarly proportioned crank trains E and C, set exactly 90 degrees apart. The motions of the two cross-heads are thus exactly proportional at all parts of their strokes. The indicator drum D is oscillated by fastening its cord to a bracket B carried on an extension of the horizontally moving cross-head. The cross-head with vertical

Fig. 67.—Error Diagram from Drum Motion Tester.

travel carries a pencil point P which traces a diagram on a card on the drum. A diagram thus obtained is one of cross-head motion shown vertically and drum motion horizontally. If a rigid connector between the drum and the bracket were used instead of the indicator cord, the motion of the drum would be exactly proportional to that of the pencil, and the diagram would be an inclined straight line. The effect of an elastic connector, as cord, variously stressed, is to give a curved line. If a straight line is drawn between the highest and lowest points of this curve, then the horizontal departure of any point on the curve from the straight line shows the error in the drum motion at that point.
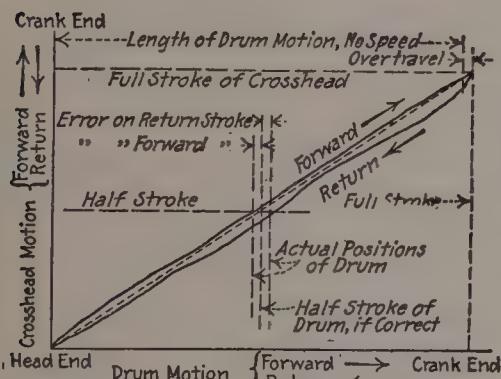
When testing a drum motion for errors in an indicator diagram previously obtained, great care should be taken to reproduce all the operating conditions, relative to speed, spring tension, length of diagram, length

of cord, general adjustment, and friction. Special care should be taken to reproduce the arrangement of guide pulleys. The effect of guide pulleys, especially if near the indicator, is similar to that of drum spindle friction and imposes an additional tightening force on the forward stroke and slackening one on the return. Fig. 67 shows a typical error diagram taken with the drum motion tester and is self-explanatory.
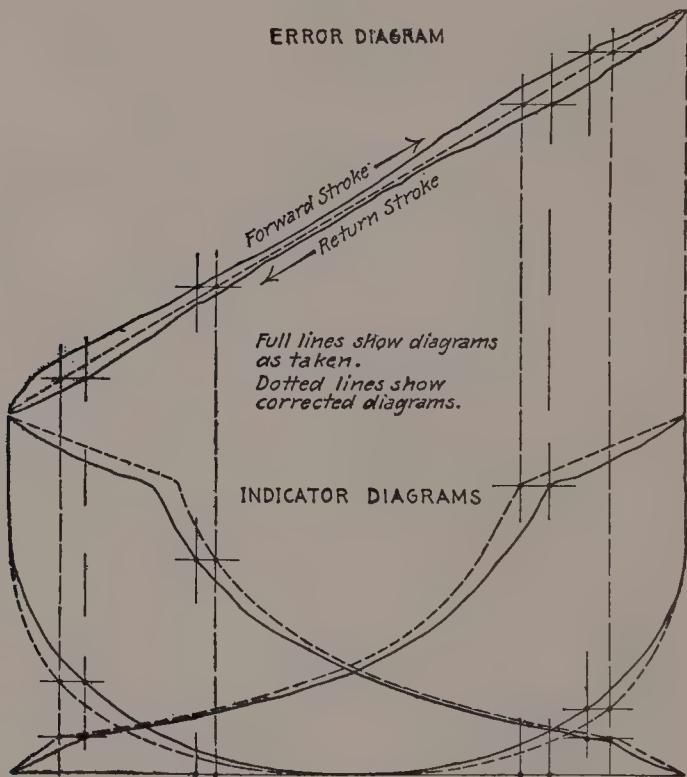
FIG. 68.—Correction of Indicator Diagrams.

(c) **The correction of indicator diagrams from drum motion tester records** may be accomplished as shown by Fig. 68, which needs no comment. Sampling of the diagrams may be done as indicated under Test 21(e).

23. THE TESTING OF LINK TYPE REDUCING MOTIONS

Principles. The pantograph in various forms has been much used to reduce engine cross-head motion for the purpose of driving indicators. Fig. 69 shows a number of them, diagrammatically. In these and in the following two figures, the letter F denotes the fixed center of the linkage;

R, the point at which the indicator cord is attached; and C, the point of attachment of the linkage to the engine cross-head.

There are two conditions necessary to the proportionality of motion of the points R and C. First, they must lie on a straight line passing through F, and second, the links that are parallel in one position must remain so in all positions.

Fig. 69. A and B are familiar forms, the reduction of motion being in the proportion of FR to FC. The identity of Fig. 69C may be established by the dotted lines, the linkage thereby represented being replaced by the sliding bar on which the point R is centered. With this arrange-
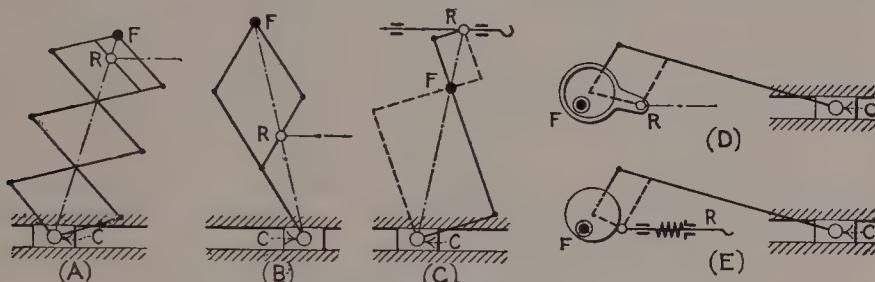
Fig. 69.—Pantagraph Reducing Motions.

ment, the ratio of the long to the short link below F must be equal to the corresponding ratio above, in order to fulfill the condition of parallelism. This reducing motion is appropriate to high-speed engines since, by it, only a short length of cord need be used. Fig. 69D is a convenient form of pantagraph made by attaching to the engine crank shaft a small eccentric and rod, or crank and rod, the throw of which is equal to the desired drum stroke. It will be seen that, for a correct reduction, the eccentricity must be in line with the engine crank; and the ratio of the lengths of eccentric rod to eccentricity must equal the ratio of the engine connecting rod to crank length. Fig. 69E is a modification of this, the motion being the same as that of a Scottish yoke, that is, a crank train with infinitely long connecting rod. The motion is therefore inaccurate.

In each case, the proportional motion of R is *parallel* to that of C. Therefore, the indicator cord should be led from the reducing motion parallel to the cross-head guides; if on a slant, the drum motion will not be proportional to the cross-head motion.

Reducing motions of the pendulum type are represented by Fig. 70. By A is shown the slotted pendulum. The shorter is the indicator cord, the greater will be its angularity due to the circular motion of R. The motion of R is not truly proportional to C since the ratio of FR to FC

varies. Fig. 70B shows the slotted cross-head, a correct motion except for the angularity of the cord. Fig. 70D is the same as C except that a "brumbo" pulley is attached, by which the cord may be led off at an angle with less error than if the pulley were not used. With a horizontal cord, the pulley causes more error than would be obtained without its use.
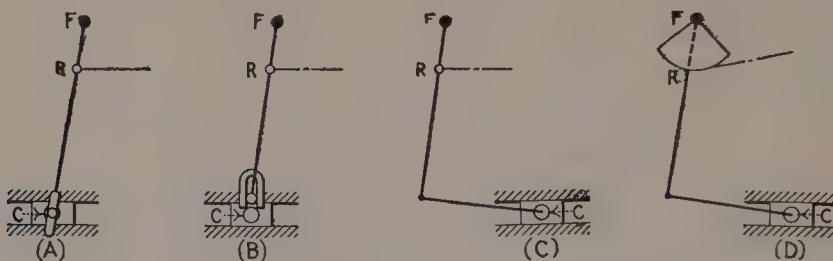
Fig. 70.—Pendulum Reducing Motions.

The errors of these reducing motions are kinematic and mechanical. Mechanical errors are due to lost motion in the joints, or flexing of the links. Kinematical truth or errors depend on the design.

(a) **Calibration by Line Diagram.** The cross-head motion is laid out to scale as shown by XY, Fig. 71, and divided into a number of equal parts. The centers of the reducing motion are then located to represent it in an extreme position, and a point D to show the corresponding position of the indicator drum. A second position of the reducing motion is then drawn and a second drum position marked, and so on. It is then an easy matter to measure the error of any of the drum positions since the distances between them should be equal for exact proportionality of motion. If desired, the data may be plotted the same as Fig. 67.

Fig. 71.

(b) **Calibration by Direct Measurement.** The indicator and reducing motion are set up as in actual use. The dead center positions of the engine cross-head are marked on the cross-head guides against a datum line on the cross-head. With the cross-head placed at any part of its stroke, its position may then be readily measured from either dead center and expressed in per cent of the stroke. For a proportional motion, the indicator drum should have moved from its corresponding dead center position (located by marking with the indicator pencil a vertical line on a card on the drum) the same percentage of its stroke. A number of
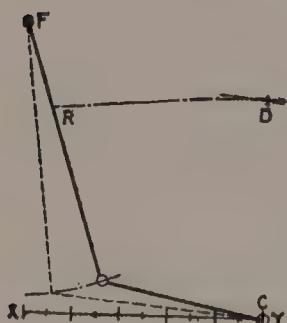
such measurements at different parts of the strokes furnish data which may be applied the same as under (*a*).

24. THE TESTING OF REDUCING WHEELS

Principles. Link reducing motions have been largely supplanted by reducing wheels on account of the latter's ready adaptability. This type of motion consists, in general, of two drums of different diameters mounted
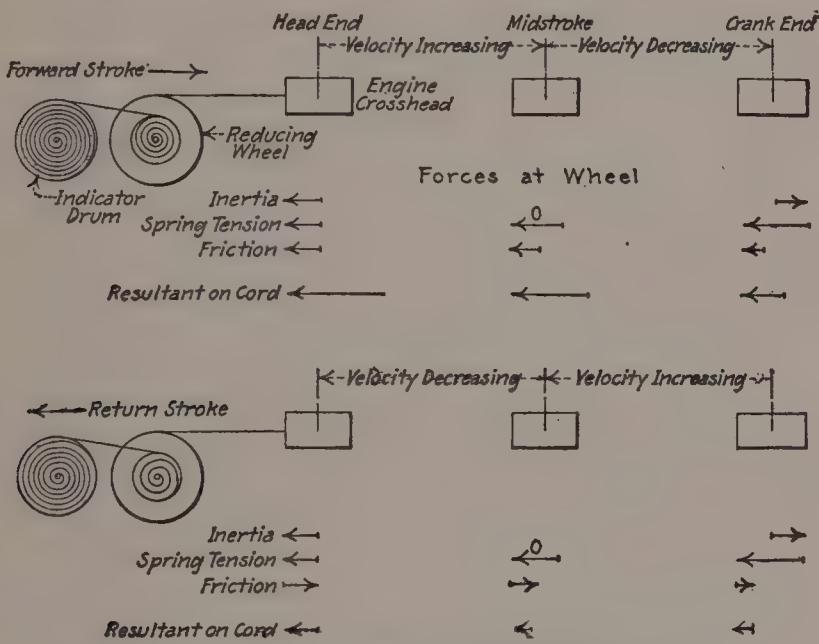
FIG. 72.—Forces on Cord Driving Reducing Wheel.

on separate shafts that are connected together with gears. Fig. 72 shows them on the same shaft for simplicity. The indicator cord is led from the engine cross-head to the larger drum to which it is fastened. Another cord connects the smaller drum to the indicator drum. It will be seen that the velocity of the one cord is to the other as the ratio of the drum diameters of the reducing motion. The reducing wheel is supplied with a spring which acts the same as the indicator drum spring. One of the reducing motion drums is made interchangeable with others of various diameters so that different reductions may be made.

In the operation of a reducing wheel the same forces are at work as in the case of the indicator drum (Test 21, principles), namely, spring tension, friction, and the force due to inertia. Since the indicator drum and wheel masses are connected by a short cord having inconsiderable

stretch, they may be regarded as one mass producing a single inertia effect. Likewise, the two springs may be considered as exerting a single force. In Fig. 72, the component forces are shown as they are felt at the cord at three parts of each forward and return stroke. The arrows represent roughly by their length the comparative magnitude of the forces.

(a) **Testing Reducing Wheels with the Drum Motion Tester.** To adapt this device, Fig. 66, to reducing wheels, a bracket, or some equivalent attachment is necessary with a long stroke to duplicate an engine
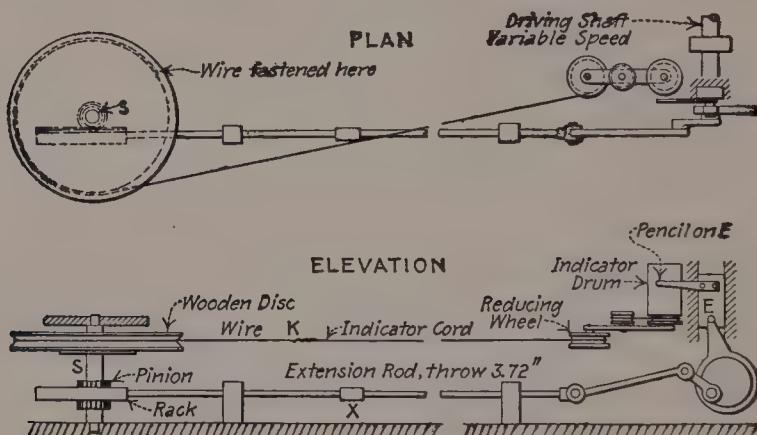
FIG. 73.—Smallwood's Drum Motion Tester Applied to Reducing Wheels.

cross-head motion; and, to make the apparatus cover all operating conditions, the stroke should admit variation. This is conveniently accomplished by coupling an extension rod to the motion tester, carrying at its outer end a rack meshing with a pinion on a vertical shaft. This shaft S, Fig. 73, carries a wooden disc replaceable by others of different diameters. When the rack is reciprocated, the disc is given an oscillating motion. An extension of the indicator cord from the reducing wheel being fastened to a point on the circumference of the disc, the cord will be reciprocated through a stroke the length of which depends upon the diameter of the disc. By using different discs, a reducing wheel may thus be operated through any desired stroke.

The reducing wheel to be tested is connected to an indicator and the whole set in place on the tester. The wheel may then be operated at any required conditions of stroke, speed, and cord length; and error diagrams similar to Fig. 67 taken. The cord length may be fixed by tying the desired length to a piece of stranded wire the other end of which is attached to the disc of the tester. The wire being practically inextensible,

the effect is to reciprocate the cord through the desired lengths as by an engine cross-head at K, Fig. 73.

Fig. 74 shows a number of error diagrams taken in this way.

(b) **Correction of Indicator Diagrams.** This may be done as described under Test 22 (c). The reducing wheel is reasonably accurate on short
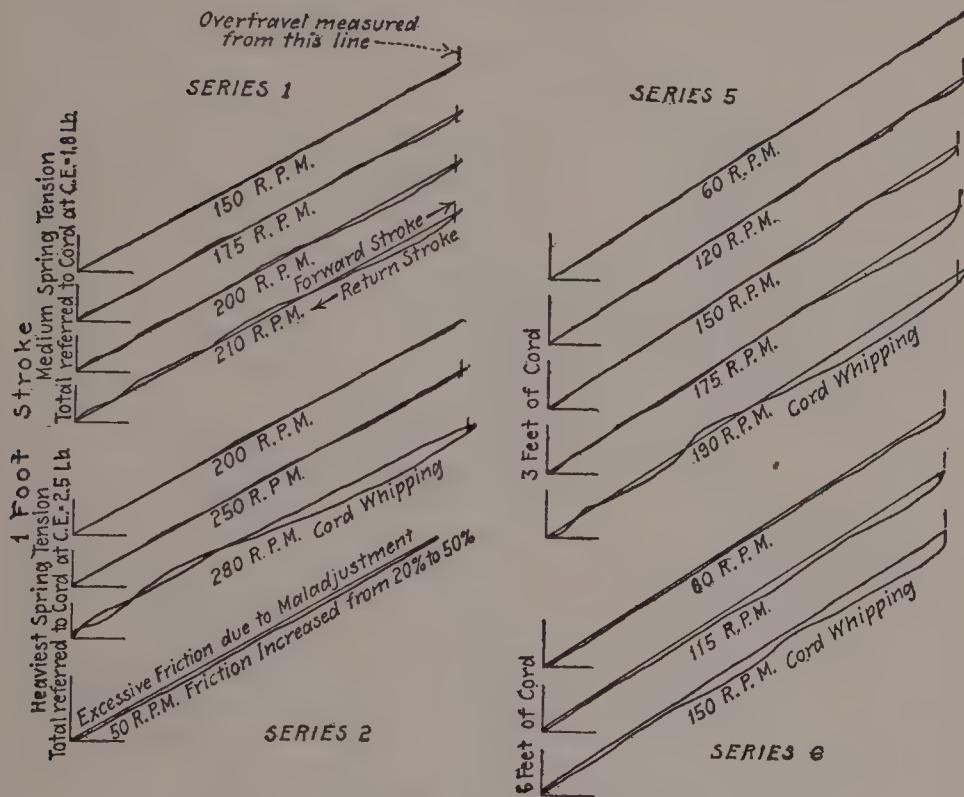
Fig. 74.—Error Diagrams from Reducing Wheels.

engine strokes and moderate piston speeds, and on long engine strokes at all usual piston speeds *if properly designed to avoid friction*. The drum motion resulting from its use has its maximum error, when the friction is small, at a part of the stroke where usually it would affect the indicator diagram little if any, namely, at the beginning of the return stroke where either the steam line on the crank end or the exhaust line on the head end diagram is described. As these lines are generally horizontal, or nearly so, errors upon them would not appear, except at their ends.

FLUID METERS

Meters for the measurement of fluids are of two broad types; those that measure weight or volume directly and those that measure velocity or rate of flow. In both cases they consist of two major elements: (1) the *primary* element which, being in contact with the fluid, is acted upon by it; and (2) the secondary element which indicates the action of the fluid upon the primary element in terms of velocity, rate of flow, volume or weight. In the simpler types, this indication is indirect, such as when a manometer indicates the difference of head existing across an orifice. The difference of head is an indication of velocity but further computation is required to determine the velocity.

Meters are available for measuring either gases or liquids but only certain types are suitable for gases while nearly all types can be adapted to measure liquids.

Weighing meters are usually in the form of a tank so arranged that, when they are filled to a certain level, they upset and spill the contents. Tanks are usually in pairs and while one is emptying the other is being filled. Another form of weighing meter consists of a pair of tanks mounted on scales so arranged that, as one becomes filled, a mechanism shifts the flow to the other and opens the valve to empty the first. The simplest form is a tank which has been calibrated in terms of the weight of fluid corresponding to various depths of liquid in the tank. Such a calibration is affected to some extent by temperature since most liquids change their density more or less with changes in temperature.

Volume meters are so arranged that all of the measured fluid passes through them, alternately filling and emptying compartments and thereby displacing a moving part which registers, through a gear and counter combination, the quantity passed. This type of meter gives the total quantity at any time. To determine rates it is necessary separately to count the time.

25. CALIBRATION OF A VOLUME WATER METER

Principles. In some types of water meter, the moving part is a piston or disc which is displaced by the water entering the compartment of which this moving part is a wall. There may be leakage past the moving part or the valve motion which controls it. This leakage will increase with the pressure drop through the meter. The greater the pres-

sure urging the water through the meter, the faster is the flow. Hence, the accuracy of a water meter will vary with the rate of flow. Other types of water meter, which may be arranged to avoid such leakage, still will have variable accuracy owing to the effect of inertia, etc., at different rates. For a complete calibration, then, a meter should be tested at enough different rates to cover its range.

(a) Calibration against a Calibrated Tank or Scales. The meter is arranged so that the water passing through it can be weighed in a tank placed on a platform scales or measured in volume from the known dimensions of the tank. If the volume is measured and the instrument reads in pounds, or vice versa, it is necessary to know the density of the water with reference to its temperature. Enough readings of both the true quantity as shown by the scales or tank and of the meter should be taken to get fair values of the rates of flow. From the observations, should be figured a series of "true rates" in pounds, gallons, or cubic feet per unit of time, and of "rates by the meter" in the same units. If desired, these may be plotted for a calibration curve.

(b) Curve of Correction Factors. This is more convenient to use with a volume meter than a calibration curve, because the instrument is not as a rule used to determine rates, but total quantities. The correction factor at any rate of flow is that number by which the total quantity as shown by the meter for any length of time is multiplied to get the true quantity. Consequently, the correction factor is the ratio of the true rate to that shown by the meter. A series of values of the factor may be calculated from the calibration curve and plotted against the rate by the meter.

In using the curve, to select the appropriate correction factor, a rough value of the rate by the meter is figured, and from the curve the corresponding factor is obtained. The true quantity for the total time is then readily determined.

26. Calibration of a Volume Gas Meter

Principles. The gasometer is the most accurate instrument of this type. It is represented by Fig. 75, and consists of two tanks as shown, the upper one being movable vertically and properly counterbalanced. This arrangement makes a chamber of variable size and water sealed. When the upper tank is raised, gas is drawn through the inlet pipe, the valve O being closed and I open. When lowered, the gas is discharged, the valve control being reversed. The volume of gas thus displaced is

measured by the vertical motion of the upper tank, its cross-section being known.

It will be noted that the gasometer cannot be used for measuring continuous flow unless two are operated, one to fill and one to discharge, alternately.

Another form of gasometer is shown by Fig. 76. Gas is drawn into the cylindrical chamber C by allowing water to flow out through the outlet pipe. This gas is then displaced through the gas outlet by causing
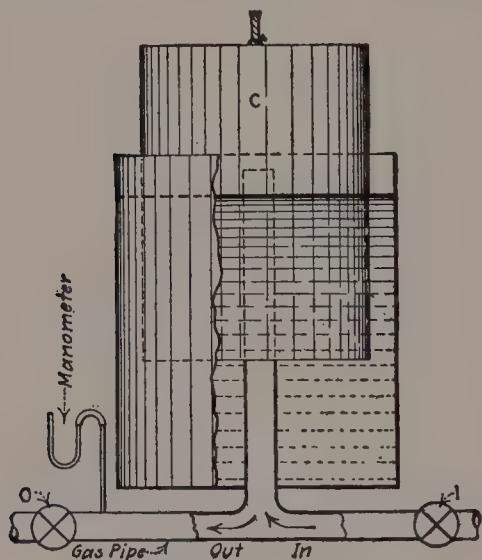
FIG. 75.—Gasometer.
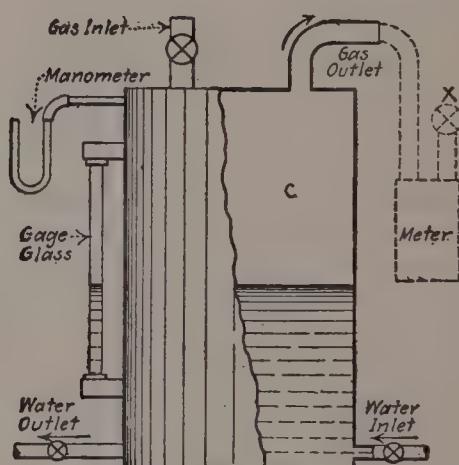
FIG. 76.—Gasometer.

water to enter through the water inlet, the valves being properly adjusted. The gas displaced is measured by the rise of water level in the gage glass.

Commercial forms of gas meter are generally of the "dry meter" type and are arranged to measure continuous flow. In this type there are two bellows chambers which are alternately filled and emptied. One side of each bellows being stationary, the other one is thus given a motion which actuates through a linkage the valve control and the recording mechanism.

As gas meters in general record volumes under the existing conditions of pressure and temperature, these conditions should be noted if it is desired to translate the readings into weights or into volumes under standard conditions.

(a) **Calibration against a Gasometer.** The meter to be tested is arranged as shown by the dotted lines of Fig. 76. The gas used for testing may be air; the gas inlet may then draw through an open pipe. The gasometer being full of air, water is caused to enter through the water

inlet, its rate of flow being the desired gas rate. This is obtained by manipulating the water inlet and noting the time of rising in the gage glass. The valve in the outlet pipe at the meter is then adjusted so as to give a constant air pressure as shown by the manometer which, if equal to that usual in city gas mains, should be about four inches of water.

For other details, see Test 25 (*a*).

(b) **Curve of correction factors** is obtained the same as for a water meter, Test 25 (*b*).

Velocity meters measure quantity by means of the relation that the flow in volume units per unit of time is equal to linear velocity multiplied by the cross-sectional area of the fluid. Such meters are dependent upon the uniformity of velocity throughout the cross-section, or upon the correctness of an estimated average velocity when it is not uniform. They may be calibrated to give volumes or weights per unit of time. To determine total quantities, it is necessary to multiply by the time.

27. Calibration of a Weir

Principles. A weir is a water meter of the velocity type. It is formed by a notch made in a dam-like obstruction in a stream of water through which all of the water is caused to flow. The level of the water behind the dam stands above the bottom edge or sill of the notch, and according to a definite difference in these levels, a definite average velocity is attained by the water, and hence the quantity passed in a unit of time. If it is arranged that the difference of level be measured by a fairly precise instrument, such as a hook gage, then the flow may be calculated; or, if the weir has been calibrated, the flow may be obtained from the calibration curve.

Fig. 77 shows a weir, the height of the water level above the sill being represented by H. The pressure to which the particles of water at the sill are

Fig. 77.—Weir and Hook Gage.

subjected is equal to H ft. of water. When this pressure is reduced to zero by emergence of the water into the atmosphere, the work done is WH ft.-lb. per sec., W being the weight of water in pounds per second. This work is transformed into kinetic energy, the expression for which is $WV^2/2g$, V being the velocity in feet per second, and g the acceleration of gravity. Consequently, $WH = WV^2/2g$ and

$$V = \sqrt{2gH}.$$
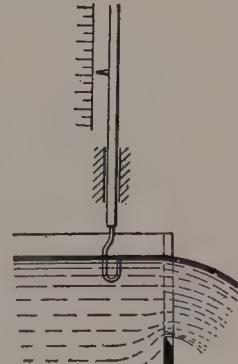
The velocity of the particles of water at levels above the sill is less than this since they are at less pressure, and it may be shown that the average velocity for a rectangular weir is $\frac{2}{3}V$ or $\frac{2}{3} \times \sqrt{2gH}$. If the breadth of the weir is B ft., then the cross-sectional area to which this average velocity applies under ideal conditions is BH sq. ft. Then, since

$$\text{Quantity} = \text{area} \times \text{velocity},$$

it follows

$$Q' = BH \times \tfrac{2}{3}\sqrt{2gH}$$
$$= \tfrac{2}{3}B\sqrt{2gH^3},$$

Q' being in cubic feet per second.

This is for ideal conditions. Actually, the velocity is somewhat less than the ideal because of eddies and friction of which the theory takes no account. So we may write

$$\text{Actual velocity} = C_1 \times \text{ideal velocity}.$$

C_1 is less than unity and is called the "coefficient of velocity."

Also the actual water section is less than BH since there is generally a contradiction due to a fall at the top and a narrowing at the edges of the weir on account of the tendency of the water at the sides to continue in the plane of the weir. Hence,

$$\text{Actual area} = C_2 \times BH$$

in which C_2 is the "coefficient of contraction."

It follows that the actual quantity

$$Q = \tfrac{2}{3}C_1C_2B\sqrt{2gH^3},$$
$$Q = \tfrac{2}{3}CB\sqrt{2gH^3},$$

C being called the "coefficient of discharge."

For rectangular weirs, C varies between 0.628 and 0.655, depending on the head. There is a tolerance of about plus or minus 3 per cent as a result of the relative sharpness of the weir crest, roughness of the weir face on the upstream side, roughness of the bottom and sides of the approach channel, etc.

When the stream behind the weir is small in cross-section, its velocity is relatively high. The so-called velocity of approach will then appreciably increase the velocity of the water through the weir since it acts in addition to gravity, and should be allowed for. The average velocity of

approach may be calculated by multiplying the velocity through the weir by the ratio of water sections at the weir and in the flume behind the weir. The additional head urging the water through the weir is then

$$\frac{(\text{Velocity of approach})^2}{2g}$$

which may be added to H for use in the formula for Q. This is an approximation since the velocity in the flume is very variable throughout the cross-section. The actual velocity of approach is greater since the water in the middle of the stream, having greatest influence on the passage through the weir, is of higher velocity than at the sides and bottom. It is more correct, then, to multiply the velocity head as just given by a number greater than unity which, according to Hamilton Smith, lies between 1.0 and 1.5.

Many modified formulas have been proposed for weirs to allow for the variations of C_2 and C. The most notable, perhaps, is Francis'

$$Q = 3.33\ (B - 0.2H)\sqrt{H^3},$$

in which 3.33 equals $C\frac{2}{3}\sqrt{2g}$. In this, it is regarded that each end contraction increases with the head and equals $0.1H$. There is thus less variation in the applied value of C. This formula may be used for uncalibrated weirs having B greater than 4 ft. when the head, H, exceeds 3 in.; with less than 1 per cent of error.

The V-notch weir is a triangular-shaped notch with the apex of the triangle down. The general equation for this type of weir is:

$$Q = \tfrac{4}{15}\ CBH\sqrt{2gH} = 2.14CBH^{\frac{3}{2}}.$$

If the apex angle of the weir is 90°, then $B = 2H$ so that the equation becomes:

$$Q = 4.28CH^{\frac{5}{2}}.$$

If the apex angle is 60°, $B = 2H \tan 30° = 1.1547H$. The equation, in this case, is:

$$Q = 2.47CH^{\frac{5}{2}}.$$

The value of the coefficient of discharge, C, varies slightly as the head is changed. In the case of the 60° notch, C decreases from 0.63 to 0.59 as the head increases from 0.2 ft. to 0.8 ft. For the 90° notch, over the same range of head, C decreases from 0.60 to 0.58 as the head is increased.

The equations take no account of the velocity of approach. To secure greater accuracy, a correction should be made by successive approximation, as follows:

First. Solve the equation, neglecting velocity of approach to obtain approximate rate of flow.

Second. Divide approximate rate of flow by cross-section area of approach channel and obtain the mean velocity of approach, in feet per second. This may be converted to a corresponding head in feet.

$$H_v = \frac{V^2}{2g}.$$

The total head, H_t, is the sum of the measured head, H, and H_v:

$$H_t = H + H_v.$$

Third. Substitute H_t in place of H in the weir equation and solve for corrected Q. One trial is usually sufficient, since the use of H instead of H_t makes only a small difference. This method is applicable to all types of weirs mentioned here.

The trapezoidal weir may be arranged so that the extra breadth due to the slope of the sides as the head rises balances the increased contraction of section and the product of the breadth B at the sill and the coefficient of contraction remains nearly constant. The slope of the sides should then be 1 to 4. The value of C may be taken as 0.63. This value will yield accurate results so long as the ratio of the head to width of the weir sill is held between 0.2 and 0.5 and the width of the sill is not less than half a foot.

The measurement of the head on a weir is accomplished usually by means of a hook gage, Fig. 77, which is capable of detecting quite minute changes in level. Such gages are usually graduated in feet and equipped with verniers reading 0.001 ft.

In use, the hook is started just below the water surface, then raised slowly, by means of the adjusting screw, until it barely breaks the surface. This can best be judged by viewing the point from an angle so that the highlight can be seen by reflected light as the point of the hook starts to break through the water surface.

The hook gage should be placed some distance back from the crest of the weir so that its readings will not be affected by the curvature of the water surface which occurs before the water passes over the weir crest. If the surface of the water is even slightly disturbed, the accuracy of the

hook gage readings will be impaired. In some cases it may be necessary to make use of a **stilling box.** This consists of a length of sheet-metal tubing or a wooden box, open at both ends, placed vertically so that its upper end comes a few inches above the water surface. Its purpose is to quell the wavelets and other turbulence. A series of screens or baffles, upstream from the weir, will aid in stilling the surface and smoothing the turbulence caused by the inflowing stream.

The zero reading of the hook gage is its reading at the exact level of the sill of rectangular and trapezoidal weirs, and of the bottom of the notch in the case of the triangular or V-notch weirs. To determine the zero reading for the rectangular and trapezoidal weirs, a straight-edge may be set with one end on the hook of the gage and the other end on the sill of the weir. A precision spirit level is placed on the straight-edge and brought to level by raising or lowering the hook. After one trial, the straight-edge and level should be turned end for end and second trial made. Care should be exercised to see that the weight of the straight-edge and level does not cause distortion of the hook and thus introduce an error because of springing the hook out of line.

When making the determination of the zero with triangular weirs it becomes necessary to go to greater trouble because the sill of the weir is nothing but a point. In this case, use is made of a round bar of known diameter. The bar is placed with one end resting in the notch and the other end supported on a small screw-jack so that it can be brought to level. The straight-edge may now be placed on top of the bar with its other end resting on the hook and brought to level as before. With the known diameter of the round bar and the angle of the notch also known, the calculation of the zero is a problem in simple geometry.

The distance from the top of the round bar to the bottom of the notch will be found to be 3.000 times the radius of the bar in the case of the 60° notch and 2.414 times the radius for the 90° notch.

To calibrate a weir, a series of readings of head and corresponding quantity rates in cubic feet per minute or second should be obtained and plotted. To determine the quantity rate, the water may be discharged into a calibrated tank, the rate of rising in which will give the volume per minute. When the weir is at the end of a flume of uniform horizontal cross-section, a convenient method is to start with it empty, and then adjust the incoming water to the desired rate of flow. The rate can be ascertained as the flume is filling by readings of the hook gage taken at equal increments of time. When the water level has reached its highest position, uniform flow through the weir being established, a

final reading of the hook gage gives the head corresponding to the rate thus previously ascertained. With this method, it is especially important that the rate be constant toward the end of each run.

28. Calibration of a V-Notch Recorder

Principles. This instrument, largely used for measuring boiler feed, is illustrated diagrammatically by Fig. 78. A float is situated in a weir

$$C H^{\frac{5}{2}} = Q$$
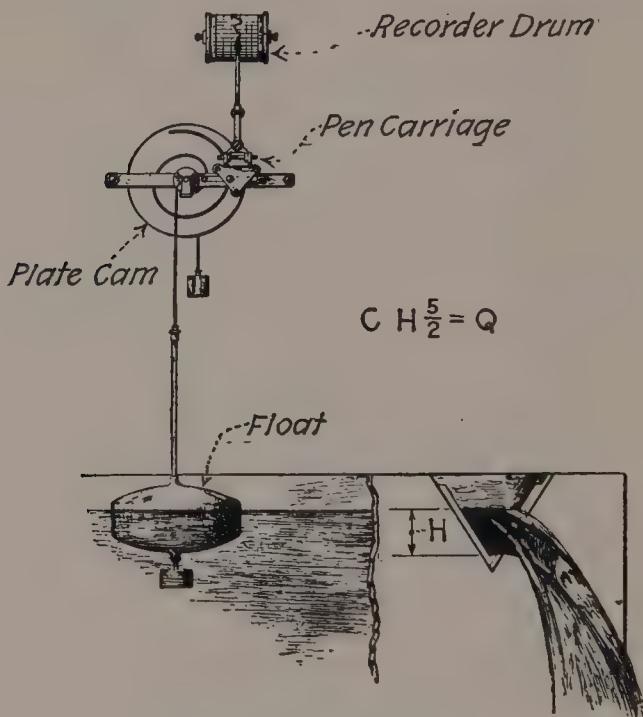
Fig. 78.—V-Notch Recorder.

tank in such a way as to be in a quiet level. A vertical spindle on the float communicates the rise and fall of the water to a plate cam by means of a steel cable wound around a drum on the cam shaft. As the cam revolves, it engages a pin on a carriage so as to move the carriage from left to right according to the height of the float, and, therefore, the rate of flow. The cam groove is a polar curve having the equation of flow; consequently the motion of the carriage is directly proportional to the flow and not to the head causing the flow. The carriage carries a pen which records on a drum chart.

The Lea V-notch recorder is much the same in general principle, but a cylindrical cam is used. Another type of V-notch recorder makes use

of a specially shaped weighing float, the vertical motion of which is proportional to the rate of flow.

The equation cited under Test 27 for a 90° V-notch is:

$$Q = 2.53H^{5/2}$$

in which Q is the cubic feet per second, and H is in feet. If h is the head in inches, and q the cubic feet per minute, then

$$q = 0.304h^{5/2}.$$

These meters are generally graduated to read in pounds, in which case there may be an error because of change of density of the water. This
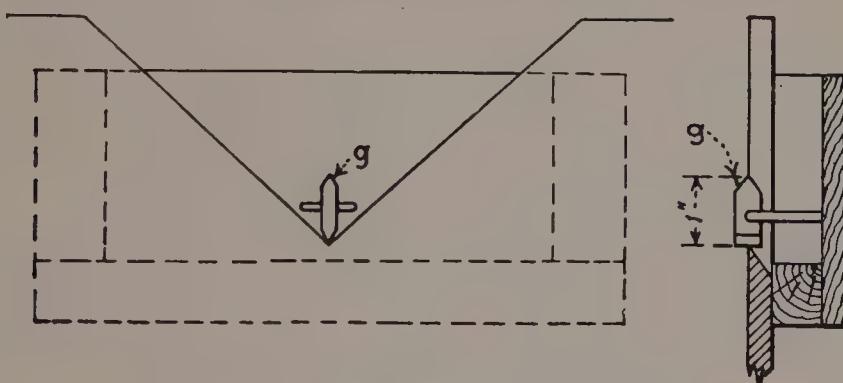
Fig. 79.—Gage Pointer for Zero Level.

error is partly compensated for, in that, at a higher temperature, for a given head, a lesser weight of water will pass, but, on the other hand, the float will stand lower in the hot water, thus making the recording pen indicate less. If the flow were proportioned to the head, the correction would be exact.

(a) **Zero Level of Weir.** Some forms of these meters are provided with an inside and outside pointer indicating this level, the outside one being adjustable. If there is none, the hook gage can be set as described under Test 27, if convenient. When the hook gage, with which the calibration is to be made, is outside the weir tank (it is often located in a vertical pipe connected at the bottom with the tank, for convenience in handling, and securing still water), the following procedure may be used.

The weir is blanked off with some boards, as indicated by Fig. 79. A gage, g, of a convenient height, say 1 in., is set in the weir as indicated. The water level should now be brought to the point of this gage, and a

reading of the hook gage taken. The zero level of the weir is then 1 in. below this reading.

(b) Calibration. Since the coefficient in the equation, $q = 0.304h$, is very well established, and is practically constant under usual operation (ranging 1 per cent above and below), this formula furnishes a ready means of calibration. It is only necessary to measure the head corresponding to any instrument record, and from this calculate the actual flow. The mechanism should first be set to register zero when under a zero head as located by method outlined in (a). Note that the height of the float should not be used for obtaining heads. Temperatures should be taken to enable the calculation of the flow in pounds.

The coefficient, 0.304, applies to a 90° notch. 54° notches also are used. This angle gives half the area of the right-angle notch. The coefficient is a little greater than half that of the latter, and may be taken equal to 0.155.

(c) Sensitiveness. With the hook gage set at a convenient height, start with a low level behind the weir, and gradually increase it until the gage is just submerged. At this instant a reading of the recorder should be noted. The procedure should be repeated, the hook gage position unchanged, with the head decreasing. The difference between the readings represents twice the lag due to friction of the recording mechanism.
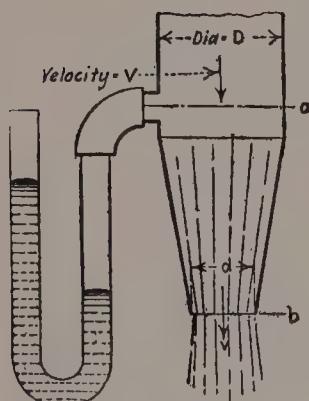
29. Calibration of a Nozzle

Principles. A nozzle makes a very convenient form of velocity water meter when the water to be measured may be discharged from a pipe into the atmosphere. The principle upon which this meter depends is similar to that of a weir, that for a definite pressure behind the opening there will be a definite flow through it. Then, if it is arranged that this pressure be measured, the flow may be ascertained in cubic feet or pounds per unit of time either by calculation or from a calibration curve.

Fig. 80.—Nozzle.

The equation of a nozzle may be obtained from a consideration of the energy appearing in the water. In Fig. 80 the energy at the section a is in two forms, pressure and velocity, neglecting the small amount of potential energy due to the height at a above the opening at b. All the energy at a is converted into velocity at b, since the pressure at b is nil,

except that used to overcome fluid friction between the points a and b. Let W equal the weight of water passing per second. Then the available energy at a is $WH + (WV^2/2g)$, H being the pressure in feet of water and V the velocity in feet per second at a. The expression $WV^2/2g$ is the kinetic energy at a, and WH the pressure energy. Likewise, the energy delivered at b is $Wv^2/2g$.

If v_1 is the velocity at b under the ideal condition of no losses, then

$$Cv_1 = v,$$

in which C is a coefficient less than unity. Since energy is proportional to the square of the velocity,

$$C^2 \times \text{available energy} = \text{delivered energy}.$$

Substituting in this the values of the energies as previously noted,

$$C^2 \left(WH + \frac{WV^2}{2g} \right) = \frac{Wv^2}{2g}.$$

We also have the relation that the velocities are inversely proportional to the cross-sectional areas, or

$$\frac{V}{v} = \frac{\dfrac{\pi d^2}{4}}{\dfrac{\pi D^2}{4}} = \frac{d^2}{D^2}.$$

Combining these last two equations we have

$$v^2 = \frac{C^2 2gHD^4}{D^4 - C^2 d^4}.$$

Since the quantity, Q, in cubic feet per second, equals the velocity times the area of the stream,

$$Q = 0.7854 d^2 \sqrt{\frac{C^2 2gHD^4}{D^4 - C^2 d^4}}.$$

For convenience, this may be expressed

$$Q = 6.3C \frac{D^2}{\sqrt{R^4 - C^2}} \sqrt{H},$$

in which R is the ratio of D to d. Note that the units of D are feet.

The value of C for a well-designed nozzle is between 0.95 and 0.99. It is thus seen that such a nozzle may be used as a meter without material error even if it is uncalibrated.

A convenient method of measuring the pressure is by a mercury manometer as shown in Fig. 80. The difference in level of the mercury in inches should then be multiplied by 13.6/12 to convert into feet of water. The lower level of mercury should be referred to the datum plane through the end of the nozzle (when the nozzle is vertical) as this allows for the potential energy between the sections a and b which was neglected in the formula; the column of water in the right-hand leg of the manometer balancing that within the nozzle. If the mercury descends below the datum plane, the reading of the manometer must be corrected for the head of water between the lower mercury level and the datum. When the nozzle is horizontal, the datum plane should be through the axis of the nozzle.

(a) **Calibration at Various Rates.** The rate may be varied by turning a stop valve in the pipe to which the nozzle is affixed, and the rate may be measured by discharging the water into a calibrated tank or by weighing it. If the pressure-measuring device is a manometer, the tube connecting it with the nozzle should be full of water as any air in it will cause the apparent water head to be different from the actual by amounts varying with the pressure. The water head below the datum plane may be corrected for by translating it from inches of water to inches of mercury and subtracting from the manometer reading; or the manometer may be raised each time it is read so that the lower mercury level is at the datum plane at the instant of reading. The determinations of rate in the desired units should be plotted against pressures in inches of mercury.

(b) **Coefficient of Discharge.** A frictionless nozzle, working under ideal conditions, would have a discharge coefficient equal to 1.00 exactly. Frictional resistance between the entrance and the throat sections tends to make the value of C less. Under abnormal conditions of operation C may become either considerably greater or considerably less than unity. With a well-made and properly installed nozzle that is of suitable size for the rates of flow to be measured, the discharge coefficient will be between 0.97 and 1.00. Under normal conditions $C = 0.985$ may be used with a fair assurance of accuracy although individual readings may vary 1 or 2 per cent. The actual value of C may be determined experimentally if the values of H, Q, and the diameters of the nozzle are known.

30. Calibration of a Venturi Meter for Water

Principles. The venturi meter is similar in principle to the nozzle. It is, in fact, a nozzle discharging into a closed and properly shaped pipe instead of into the atmosphere (see Fig. 81). The water passing through the pipe shown carries a certain amount of energy in the form of pressure and velocity. When the water reaches the contracted section b, its velocity is increased, and therefore its pressure must be decreased, since the total energy, barring losses, remains constant. The drop in pressure between sections a and b thus becomes a measure of velocity and hence quantity.

To deduce the equation for the venturi meter it is only necessary to equate the expressions for energy at the two sections a and b. Thus, if V and v are the velocities in feet per second, and H and h the pressures in feet of water at the sections a and b, respectively, and W the weight of water passing per second, then

Fig. 81.—Venturi Meter.

$$WH + \frac{WV^2}{2g} = Wh + \frac{Wv^2}{2g},$$

g being the acceleration of gravity. We also have the relation

$$\frac{V}{v} = \frac{\text{Area at } b}{\text{Area at } a} = \frac{d^2}{D^2}.$$

Combining these equations and solving for v,

$$v = \frac{D^2}{\sqrt{D^4 - d^4}} \sqrt{2g(H - h)},$$

from which the cubic feet per second discharged under ideal conditions is
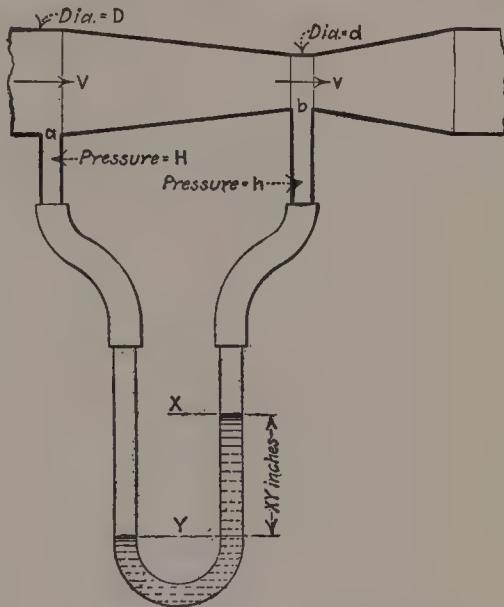
$$Q' = \text{area} \times \text{velocity}$$

$$= 0.7854 d^2 \frac{D^2}{\sqrt{D^4 - d^4}} \sqrt{2g(H - h)}.$$

Owing to friction losses the discharge thus calculated is too high. To allow for this, a coefficient of discharge, C, less than unity, is introduced. Combining $\sqrt{2g}$ and 0.7854, for convenience, the formula becomes

$$Q = 6.3C \frac{D^2}{\sqrt{R^4 - 1}} \sqrt{H - h},$$

in which R is the ratio of D to d, and D is in feet.

The theoretical equation may be expressed in another form:

$$Q = \frac{A}{\sqrt{R^4 - 1}} \sqrt{2gH}$$

where A is the area of the entrance section of diameter D. The portion of the equation

$$\frac{A}{\sqrt{R^4 - 1}}$$

is called the meter constant, M, and may be computed, once for all, from the measured dimensions of the venturi. The theoretical equation may then be written:

$$Q = M\sqrt{2gH}.$$

This form of the meter constant is sometimes criticized on the ground that it overemphasizes the importance of the entrance section, A. The throat area, a, corresponding to the diameter, d, is also an important factor, since any change in the orifice area has a decided effect on the value of the equation. As an alternative, the following may be used:

$$Q = a \sqrt{\frac{r^2}{r^2 - 1}} \sqrt{2gH}$$

where

$$r = \frac{A}{a} = \frac{\text{entrance area}}{\text{throat area}} = \frac{D^2}{d^2}.$$

This alternative form has the advantage of involving only squares and square roots. The practical form of this equation is

$$Q = Ca \sqrt{\frac{r^2}{r^2 - 1}} \sqrt{2gH}.$$

The connection to the pressure-measuring device may be taken through a single small hole drilled perpendicularly through the wall of the venturi at the throat and entrance. The theory is that if the hole is too small to create any sensible disturbance in the motion of the fluid along the wall, the motion of the fluid along the wall will have no effect on the pressure observed. This method of pressure measurement seems to be justified by the results since values so obtained, when used in theoretical equations as if obtained in strict accordance with the definition of static pressure, do not exhibit any conflict between theory and experiment which can be attributed to error in the principle of measurement.

Venturi meters and similar devices are frequently equipped with a number of small holes leading into an annular pressure chamber called a "piezometer ring" (see Fig. 82). The idea of the piezometer ring is to reduce any errors which might be caused by perma-

FIG. 82.
Piezometer Ring.

nent eddies or cross currents acting on a single hole. It is commonly supposed that the piezometer ring is the most reliable means of determining static pressure. It is doubtful whether this supposition is well founded, for if the pressures at the separate holes are appreciably different, there must be some circulation through the ring. If this is true, then there is no reason to suppose that the single gage connection will give the arithmetical mean of the pressures at the individual holes.

Furthermore, if the pressures at the individual holes are appreciably and permanently different, assuming the holes to be properly made, it is thereby demonstrated that the fluid is not flowing straight along the pipe but has permanent cross currents and eddies. The theoretical equations are all based upon straight flow and they cannot be expected to accurately represent the facts when other than straight flow is experienced.

For measuring the difference of static pressure, $H–h$, a mercury manometer is generally used connected as shown by Fig. 81. When the manometer is graduated in inches, the flow may be calculated or obtained from a calibration curve determined experimentally. The manometer is sometimes graduated in quantity rates so that no curve is necessary.

Some forms of venturi meter employ apparatus giving a continuous record on a time chart which, when integrated, yields total quantities.
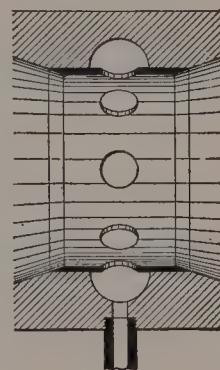
One of these is illustrated by Fig. 83. Note the cam by which movement of the mercury column (proportional to the square of the velocity) is rectified to give uniform radial chart ordinates.

If reliable results are to be expected from a venturi meter it is essential that the meter should not follow immediately after an elbow, valve,
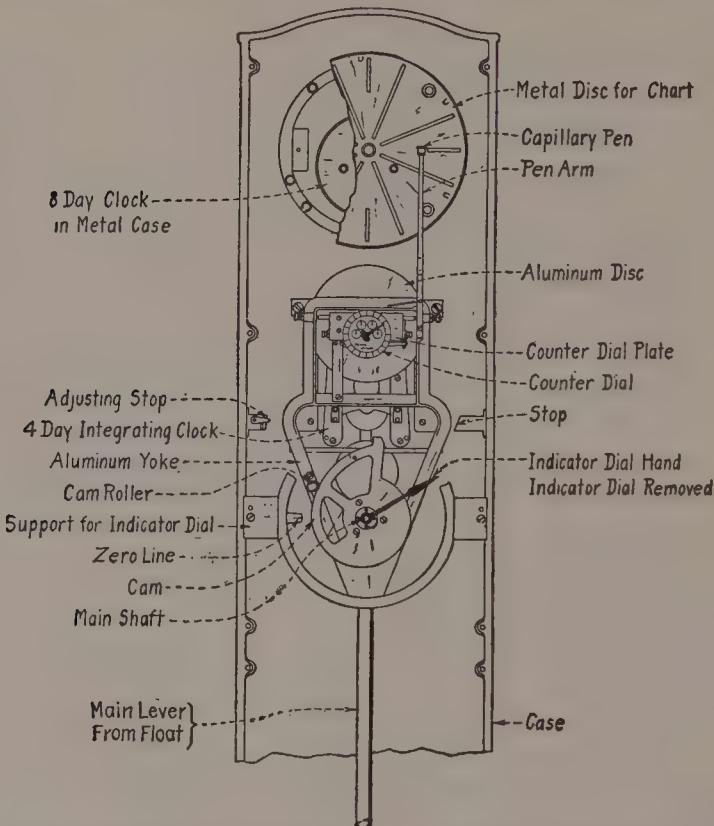
FIG. 83.—Builder's Iron Foundry Venturi Meter Recorder.

or other irregularity in the pipe line which would tend to cause turbulence or set up permanent cross currents and eddies. It is recommended that a length of at least 5 diameters of straight pipe precede the meter; more is better but there is nothing to be gained by having more than 20 diameters of straight pipe preceding the meter.

When it is impossible to provide a sufficient length of straight pipe before the meter, the flow should be steadied by placing a sheet metal cross, at least 2 diameters long, in the pipe just ahead of the meter.

(a) **The Calibration at Various Rates** may be made by controlling the rate by a stop valve in the water line and measuring the rate by discharging the water into a calibrated tank or by weighing it. A curve

should be drawn between the rates and the differences of mercury level (manometer readings).

The head in inches of mercury should be transposed to feet of water. Referring to Fig. 81, it is seen that the column of water on the right above X balances an equal column on the left. The column of water between X and Y, however, is balanced by mercury, so that the pressure difference between sections a and b is not XY inches of mercury, but XY inches of mercury minus XY inches of water; that is

$$\frac{(13.6 - 1)}{12} XY = 1.05 XY, \text{ feet of water.}$$

(b) **Coefficient of Discharge.** The value of C for a well-designed venturi meter lies between 0.92 and 0.99. The smaller venturi meters show the greatest variation in the value of C for different rates of flow; the smaller values of C being associated with the lower rates of flow. With very large venturi meters, the value of C becomes practically a constant over the entire working range except at throat velocities around 1 or 2 ft. per sec.

The coefficient may be determined experimentally for any rate when the values of Q, $H-h$, and the diameters of the instrument are known. The value of R is usually made 2 or 3 to 1.

31. Calibration of a Venturi Meter for Gas

Principles. Fluid meters were originally invented for the measurement of liquids, but they are equally applicable to the measurement of gases and vapors without any change in the design of the meter. This applies not only to venturi meters, but also to nozzles and orifices. The static pressure, set up by the constriction of the stream, cannot be measured directly in head of fluid flowing but must be measured indirectly by means of liquid manometer or other device.

The deduction of the flow formula differs from that for water on account of the fact that gases and vapors carry intrinsic energy, due to their expansive properties, which must be accounted for in the energy equation. Such treatment, however, leads to a very complicated equation which is inconvenient for purposes of computation. In the A.S.M.E. Research Report on Fluid Meters, pages 46–50 (Fourth Edition, 1937), there is developed a method of using and applying correction factors to the hydraulic equation similar to that set up for water. The correction fac-

tors depend on (*a*) the diameter ratio, R, (*b*) the ratio of differential to initial pressure, and (*c*) the specific heat ratio of the gas or vapor being measured. The correction factors have been worked out in tabular form which greatly shortens the work of computation.

(**a**) **Calibration at various rates** may be accomplished by using one of the methods described under Tests 33, 35, 36, or 37 for measuring the true quantity of gas; or if the venturi meter is of small size, by using a gasometer. The rates may be obtained in cubic feet per second or minute, and plotted against difference of pressure in inches of water or mercury. Note that the temperature conditions must not materially vary from those in use, since a change of temperature is accompanied by a change of flow.

32. Calibration of a Pitot Meter for Water

Principles. Fig. 84 represents a Pitot meter, which is merely a curved tube placed in a stream of velocity V feet per second so that the immersed end of the tube faces the stream. The kinetic energy of a given weight of the water, W, is $WV^2/2g$, in which $V^2/2g$ is called the "velocity head" or the head of water, H, in feet, which under ideal conditions may produce a velocity, V. In the Pitot tube, Fig. 84, the water rises until the head in the tube just balances the velocity head of the stream. Hence

$$V = \sqrt{2gH}.$$

The height H thus becomes a measure of velocity and therefore quantity.

If the stream is in a closed conduit, the water may be under a pressure greater than atmospheric which would cause it to rise in the Pitot tube to a height greater than that due to velocity. This may be allowed for by making a separate measurement of pressure. Thus, in Fig. 85, H_2 feet of water balances velocity plus pressure, and H_1 feet in the straight tube is due to pressure only, owing to the fact that the velocity is not impressed upon this tube opening. The velocity is then

$$V = \sqrt{2g(H_2 - H_1)}.$$

The Pitot tube opening is called the "velocity opening" and the other, the "static opening." They are often connected to a differential gage as shown by the dotted lines of Fig. 85 so that the pressure is balanced, and a single reading gives the velocity head direct.

The plane of the static opening should be parallel to the stream; for, if it is inclined toward it, the velocity is partially impressed; or, if away from it, suction will result. With a correct static opening there may be the same effects if the stream is not parallel to the pipe direction, which may be the case at points near bends or elbows, or when the instrument itself interferes with the regularity of the flow. Generally two static openings on diameters at right angles, connected together, are used.
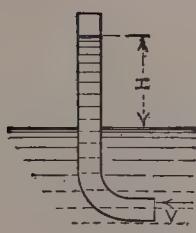
FIG. 84.—Pitot Tube.
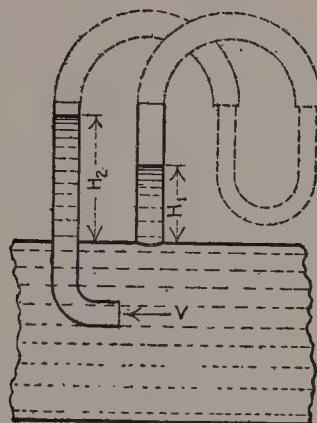
FIG. 85.—Pitot Meter.

When the Pitot tube and static opening are properly designed and applied, the actual condition of flow is the same as that represented by the formula so that there need be no coefficient to determine as with the nozzle, weir, and venturi meters.

In a closed conduit, the velocity of the stream varies through the cross-section, being maximum at the center and minimum at the sides. It is therefore necessary either to find the velocity at enough points for a fair average or to read the instrument at a point of mean velocity. The mean velocity is approximately $0.83 \times$ velocity at center, and this rule is sometimes used for rough results.

(a) **Determination of Average Velocity and of Quantity Rates by Traversing.** In Fig. 86, the concentric circles mark off equal areas so that $a_1 = a_2 = a_3 = a_4 = a_5$, the area of the pipe being $A = 5a_1$. The average velocities in these areas are v_1, v_2, v_3, respectively, the average velocity in the whole pipe section being V. Then, if Q represents the cubic feet per second,

$$Q = AV = a_1v_1 + a_2v_2 + a_3v_3 + \text{etc.},$$
$$= 5a_1V = a_1(v_1 + v_2 + v_3 + \text{etc.}),$$

and

$$V = \frac{v_1 + v_2 + v_3 + \text{etc.}}{5}.$$

That is, the average velocity in the pipe is the average of velocities from points representing equal areas.
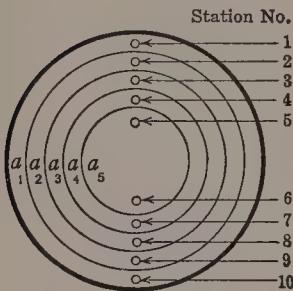
The method to be pursued, then, is to take readings of the differential gage at such points for a single determination of quantity, the quantity being calculated from the product of the average velocity, V, and the area of the pipe. It is customary to take readings at ten points, or stations, as shown by Fig. 86. The readings on one side of the pipe center should duplicate those on the other, and there are two readings for each area. The stations should be located at the following distances from the pipe wall, D being the diameter of the pipe, in order to represent equal areas.

FIG. 86.

Station No. 1,	0.0256D	No. 6,	0.658D
No. 2,	.0817D	No. 7,	.774D
No. 3,	.146D	No. 8,	.854D
No. 4,	.226D	No. 9,	.918D
No. 5,	.342D	No. 10,	.974D

The calculation for V is simplified by taking the average of the square roots of the differential gage readings, and using this in the velocity formula for $\sqrt{H}$. Note that the head of the gage liquid should be converted into equivalent head of water in feet. If mercury is used as the gaging liquid, this may be done as described under Test 30(b). If some other liquid, as oil, is used, its specific gravity should be accounted for according to the same principle. The quantity rate can be expressed as

$$Q = \text{a constant} \times \Sigma\sqrt{\text{gage reading}}$$

for the simplification of numerical work.

Recording Pitot meters in various forms are in considerable use, and of these a good example is that of the General Electric Company, applicable also to steam and gas flow. These instruments are calibrated with the velocity opening at a fixed position in the stream, and yield charted results in terms of pounds, gallons and cubic feet per unit of time.

(b) Location of the Point of Mean Velocity. If the pipe is traversed and the average of the square roots of the gage readings obtained as under (a), then the square of this average represents the gage reading at the point of mean velocity. This point may then be located by plotting on a chart gage readings against distances from the wall of the pipe; the distance corresponding to the square of the average square root of the gage readings being then taken from the chart.

The location may be found without plotting a curve by searching with the Pitot tube for the point at which is registered the gage reading as calculated and then measuring the distance of the velocity opening from the pipe wall. If this is done, a center reading should be taken when making the traverse by which the uniformity of flow may be checked when the searching is done.

33. CONSTANTS OF A PITOT METER FOR GAS

Principles. The Pitot meter may be used for gas in exactly the same manner that it is used for water (see Test 32) except that greater precautions should be taken on account of the fact that gas is a much more mobile fluid. The instrument should be placed at a distance of at least 12 pipe diameters from the nearest bend on the upstream side, and four on the downstream. A Pitot tube of unusual proportions or design should not be used without previous calibration, as it has been found by experiment that apparently unimportant details cause large errors in the

FIG. 87.—Pitot Tube for Air.

indications. The best proportions of Pitot meter for gas under general conditions have not yet been satisfactorily established, but Fig. 87 represents the form recommended by the American Society of Mechanical Engineers. Numerous experiments, however, lead to the conclusion that it is sufficient to have two static openings at the wall of the pipe, instead of as shown by Fig. 87.*

In the case of gas, in the formula

$$V = \sqrt{2gH},$$

H is the head due to velocity expressed in feet of whatever gas is flowing, allowing for its density due to its condition of pressure and tempera-
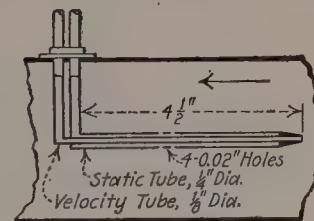
* See work of William Rawse, Vol. 35, *Trans.* A.S.M.E.

ture. The observed head in terms of inches of the gaging liquid must therefore be translated. For this purpose the following relation may be used,

$$H = \frac{h}{12} \times \frac{\text{density of gaging liquid}}{\text{density of gas in pipe}},$$

in which h is the observed head in the U-tube connected as shown in Fig. 85, expressed in inches. The densities are generally given in pounds per cubic foot. Water is usually used for the gaging liquid, the density of which may be taken as 62.3 lb., as it varies but little with temperature. Kerosene is also used. Its density may be figured from its specific gravity.

(a) **Determination of Average Velocity and of Quantity Rate by Traversing.** The experimental procedure is exactly the same as for Test 32(a) when the gas is at room conditions of temperature and pressure, that is, approximately 70° F., and 14.7 lb. Air, for example, under these conditions weighs 0.0749 lb. per cubic foot, so

$$H = \frac{h \times 62.3}{12 \times 0.0749} = 69.3h,$$

from which

$$V = \sqrt{2g(69.3h)} = 66.7\sqrt{h},$$

in which h is the observed head in inches of water.

At other pressures and temperatures, the density varies according to the relation (for perfect gases)

$$144pv = p\frac{144}{w} = RT,$$

in which p = absolute pressure in pounds per square inch;
v = specific volume in cubic feet per pound;
w = density, in pounds per cubic foot;
R = 53.4, for air;
T = absolute temperature, degrees F.

From this relation, follows

$$w = \frac{144p}{RT}.$$

Now

$$V = \sqrt{2gH} = \sqrt{2g\frac{h}{12} \times \frac{62.3}{w}}.$$

Substituting the value of w and simplifying,

$$V = 11.1 \sqrt{\frac{h}{p} T},$$

from which the quantity, Q, in cubic feet per second, may be figured by multiplying by the area of the conduit.

If the quantity rate, in pounds per second, W, is wished, the following may be used:

$$W = wQ = \frac{144p}{RT} \times 0.7854 \frac{d^2}{144} \times 11.1 \sqrt{\frac{h}{p} T},$$

in which d = diameter of conduit in inches. Simplifying,

$$W = 0.163d^2 \sqrt{\frac{hp}{T}}.$$

The absolute temperature is computed by adding 460° to the temperature as obtained by a Fahrenheit thermometer. For the absolute pressure of the gas, p, the barometer should be read and added in the same units to the static pressure of the gas as shown by a U-tube or other pressure gage. This gage may be connected to the static opening of the Pitot meter.

For any other gas than air, or for any other gaging liquid than water, an equation may be deduced similarly by the same method.

The procedure, then, for gas at temperatures and pressures materially different from room conditions is either

First, to make a traverse from which the average of the square roots of the differential gage readings yields $\sqrt{h}$, and to get one set of readings of temperature, static pressure, and barometer.

Second, to place the Pitot tube at a point of average velocity, by which h is obtained by a single reading, other measurements the same.

(b) **The Location of the Point of Mean Velocity** for gas may be found in exactly the same manner as that for water, Test 32(b).

34. CALIBRATION OF AN ORIFICE FOR WATER

Principles. If water is caused to pass through an orifice, the pressure behind the orifice being measured, the flow may be calculated. Suppose that the velocity of the water behind the orifice is negligibly small; then the energy available is WH, W being in pounds per second, and H, the

pressure in feet of water. This pressure energy is expended in imparting velocity, or kinetic energy, to the water upon passing through the orifice, so that, neglecting friction, etc., $WH = WV'^2/2g$, V' being the velocity in feet per second, and g the acceleration of gravity. Hence, under ideal conditions, $V' = \sqrt{2gH}$. Owing to friction losses, the actual velocity is somewhat less than this, so that if c_1 is a number less than one,

$$V = c_1\sqrt{2gH}.$$

Multiplying this by the cross-sectional area of the stream gives the quantity, Q, in cubic feet per second. Now, this area is less than that of the orifice, the stream being contracted by virtue of the tendency of the water particles to flow in the plane of the orifice; so that if c_2 is a number less than one,

$$\text{area of stream} = c_2 \times 0.7854d^2,$$

the orifice being circular and d feet in diameter. It follows that

$$Q = c_1c_2 \times 0.7854d^2\sqrt{2gH}$$

$$= c \times 0.7854d^2\sqrt{2gH}.$$

c_1, c_2, and c are called the "coefficients of velocity, contraction and discharge," respectively. Merriman's values for them are 0.98, 0.62 and 0.61 for the case of orifices with sharp edges, that is, orifices in thin plates beveled on the side not touched by the water. In this Merriman assumes the velocity of approach to be negligible and the discharge passing into atmosphere.

Where such an orifice is installed in a pipe line, the velocity of approach cannot be neglected. Furthermore, the position of the pressure taps, with respect to the orifice, has a decided effect, particularly true of the downstream tap. Since the velocity of approach is related to the ratio of orifice diameter to pipe diameter, the coefficient of discharge will vary with this ratio.

When a fluid flows through a sharp-edged orifice, the stream contracts to a minimum diameter at a point some distance downstream from the orifice plate. The point of minimum diameter is known as the *vena contracta* and is the point of highest velocity and lowest pressure; if the pressure tap is connected to the pipe, opposite the vena contracta, the greatest difference of pressure, across the orifice, will be realized. However, turbulence is also greatest at this point and unsteadiness of manometer indications will result.

The A.S.M.E. Fluid Meters Report * gives very complete data on the values of the coefficient of discharge for sharp-edged orifices. This publication should be consulted when designing or selecting an orifice for a given installation.

Orifices which are well-rounded on the upstream face do not exhibit the marked contraction (vena contracta) as is the case with the sharp-edged orifice. Such well-rounded orifices are generally classed as flow nozzles and certain standard forms have been developed. As might be expected, the coefficient of discharge is higher for the well-rounded type than for the thin plate, sharp-edged, orifice; it is usually on the order of 0.95, but variations will be found depending on size and shape of the flow nozzle.

(a) **Determination of quantity rates at various heads** may be made by discharging the water into a weighing tank, or by measuring its volume. The quantity discharged in cubic feet per minute should be plotted against the head in the observed units.

(b) **Determination of Coefficients.** The value of c may be calculated from the formula when the values of Q, H, and d are known. The coefficient of contraction may be found by measuring the diameter of the stream at the contracted section with a caliper, this section being distant from the plane of the orifice by about a half diameter. Knowing this coefficient, and the coefficient of discharge, the coefficient of velocity is readily obtained. This can be done, however, only with orifices discharging into the atmosphere.

35. CALIBRATION OF AN ORIFICE FOR GAS

Principles. The deduction of the flow formula differs from that for water on account of the fact that gas carries intrinsic energy due to its expansive property, which must be accounted for in the equation of energy. The theoretically correct flow formula is deduced in various works on thermodynamics. It includes necessarily a coefficient c to allow for contraction and losses, as is the case with water passing through an orifice.

When the pressure drop between the two sides of the orifice is small, the volume of the gas changes but little, and consequently only little intrinsic energy is delivered to influence the flow. Under these circumstances, the intrinsic energy may be ignored, and an equation deduced

* Fluid Meters, Their Theory and Application, American Society of Mechanical Engineers, 29 West 39th St., New York, N. Y.

similar to that for water, namely, $V = c \sqrt{2gH}$, the head, H, then being the height of a column of gas to produce the pressure drop, its density being taken into account.

Since the orifice generally may be designed of such size to produce a small pressure drop, it seems hardly worth while to use the accurate and much more complex formula, especially in view of the uncertainty of the values of the coefficient of discharge under those conditions to which the hydraulic formula does not apply.

If the orifice is to be installed in a pipe line in such a manner that the velocity of approach and the compressibility of the gas cannot be ignored, the more complex equations must be employed. The derivation of these equations is too lengthy for inclusion here but may be found in the standard works on thermodynamics or in the A.S.M.E. Fluid Meters Report previously mentioned in this section.

(a) Determination of Quantity Rates at Various Heads. The rate may be changed by varying the output or the speed of the machine handling the gas. If an air compressor or blower, the speed may be varied. If the orifice is used for such a purpose as measuring the exhaust from a gas engine, its external load may be varied, thus changing the amount of fuel and air used.

The true rate may be measured by any of the methods of Tests 33, 36, 37 or by a gasometer if the volume of gas is not too large.

Readings of the pressure difference between the two sides of the orifice should be plotted in the gage units against true rates.

(b) Determination of the Coefficient of Discharge. This is readily calculated from the flow formula, the calibration data of (a) being known. It is instructive to plot values of c against the pressure drop through each size of orifice, and against orifice diameters for each value of the pressure drop.

36. Calibration of an Anemometer

Principles. An anemometer is a meter of the velocity type, generally consisting of a wheel with vanes against which the current of gas impacts causing a rotary motion proportional to the velocity of the current. This motion is transmitted to a gear and counter combination which registers linear feet continuously. By counting the time, the velocity in feet per second or minute is calculable.

When used to measure quantity, the anemometer is generally placed at the exit cross-section of the conduit discharging the gas. If the cross-section is rectangular it may be divided into a number of small squares

defined, for convenience, by light strings or wire fastened from wall to wall of the conduit. The anemometer is placed in the middle of each of these squares and the velocity read. The average velocity through all of them may then be used with which to multiply the total area to get cubic feet per minute or second. If the conduit is round, the anemometer should be placed at points located as for a Pitot tube, Test 32(a).

Anemometers generally are not adapted for velocities higher than 100 ft. per sec., and are not very reliable. The vanes are apt to become deformed, causing false indications, and changes of frictional resistance of the bearings will have the same result.

When the velocities exceed the capacity of the anemometer, the discharge conduit may be enlarged in cross-section at the exit.

(a) Calibration Against the Velocity of the Instrument in Still Air. The anemometer is mounted on one end of a horizontal arm 3 or 4 ft. long and pivoted at the other end causing the instrument to travel through the circumference of a circle. For this purpose the pivot may be supplied with a grooved wheel whereby the arm may be driven, through a belt, by a motor or by hand. Knowing the revolutions per minute of the arm and its radius to the center of the anemometer, the linear velocity of the anemometer may be calculated. This is the true velocity of the air relative to the anemometer, and corresponds to the instrument reading. A number of such determinations are made at different velocities, and plotted as a calibration curve.

Care should be taken that the velocity is uniform throughout each trial, and that the error of starting and stopping is sufficiently small. A small lever is generally arranged on anemometers by which the recording mechanism may be thrown in or out of gear. This may in some cases be operated while the instrument is moving.

(b) Curve of Correction Factors. Corrections may be figured as quantities in linear feet per minute to be added to or subtracted from the instrument indication for one minute. These should be plotted against linear velocities as shown by the instrument.

37. Testing a Calorimetric Apparatus for Measuring Gas

Principles. If gas flowing through a conduit is arranged to be heated or cooled by steam, water, or electric current, then, barring radiation from the conduit, the heat gained by the one medium equals that lost by the other. Thus, supposing the gas to be cooled by water pipes, if

W = weight of gas passing in a given time;

W_w = weight of water passing in same time;

T_1, T_2 = initial and final temperatures of the gas;

t_1, t_2 = initial and final temperatures of the water;

C_p = specific heat of the gas at constant pressure;

then

$$WC_p(T_1 - T_2) = W_w(t_1 - t_2),$$

from which,

$$W = \frac{W_w(t_1 - t_2)}{C_p(T_1 - T_2)}.$$

It is thus seen that with such an apparatus the air passing in a given time may be measured by weighing the water and taking the temperatures of the gas and water before and after cooling.

If steam is used in the coils, being condensed by the air, it is necessary to take the temperature of the water discharged and the pressure and quality of the entering steam from which its heat content may be obtained with the steam tables.

If electric current is used, the heat equivalent may be figured from voltmeter and ammeter readings.

The calorimetric method is sometimes useful for measuring large quantities of air or gas.

The Thomas Electric Gas Meter is an elaborate apparatus of this type, and is perhaps the most accurate device on the market for measuring gas, especially in large quantities. Resistance thermometers are used, by which a constant temperature difference is maintained; and the current necessary to maintain this difference of temperature is measured in terms of standard cubic feet of air. The meter is entirely automatic and autographic.

An advantage of this type of meter is that no corrections for pressure and temperature are necessary, since the indications are proportional to weight and, therefore, to standard cubic feet.

(a) **Examination of Instruments.** The instruments should be sufficiently precise that the error of reading should be less than 2 per cent of the corresponding factor in the formula. The weight of water may be measured readily with proper precision. The temperatures, however, appear as differences which may be only a few degrees. Hence, thermometers graduated to tenths may be necessary. It is sometimes useful to figure beforehand rough values of weights and temperature differences in order to ascertain the required precision of the instruments.

When using the apparatus, it is best to take temperatures at various points in the cross-section of the gas conduit, to search for variations.

(b) Determination of radiation correction may be made approximately by stopping the flow of gas and maintaining the temperature within the conduit at an average value between the limits in actual operation. The heat necessary to maintain this temperature may then be figured for a unit of time. This may be used as a correction if very precise results are desired.

38. Calibration of a Steam Meter

Principles and Types. Steam meters are built on the Pitot, venturi, and orifice principles. The quantity of steam flowing, in pounds per unit of time, is therefore proportional to the square root of a pressure or difference in pressure. The differential pressure, varying with the flow, actuates an indicating or recording device. When arranged to give a time chart showing rates and total quantities, the recording mechanism is generally rather delicate and complicated. It should be noted that with such meters, the steam flow is proportional not to the motion of the indicator or height of the chart, but to the square root of these quantities, unless some rectifying device is employed.

Changes in density of the steam, due to difference in pressure or superheat, should be allowed for, since the meters generally indicate the quantity in terms of pounds. This may be done by applying different tables, furnished by the makers, to interpret the indications, or by a hand adjustment of the recording mechanism. Variation of density due to wetness may be avoided by passing the stream through a separator just before it reaches the meter.

Steam meters generally are not to be depended upon when the flow is variable or intermittent. This condition may often be remedied by the location of the meter.

Integration of drum charts with uniform ordinates may be accomplished with the usual polar planimeter; but for circular charts the methods of Test 15 (*b*) may be applied.

(a) Calibration may be made by comparison with an accurate steam meter, or by direct weighing of the steam. The latter method is the more dependable. All of the steam passed through the meter in a given time should be condensed in a surface condenser and then weighed. A calibration curve may be made for each pressure or condition of superheat, if desired, the quantity of steam being varied by a valve in the

steam line, between the meter and condenser. Generally it is sufficient to make the calibration at one condition of pressure, the steam being saturated. Throttling calorimeter readings should be made to insure the latter condition. The rate of flow should be kept as constant as possible during each of a number of runs at different rates, and the total condensate compared with the total obtained by integration of the chart, or average indications multiplied by time.

Before testing, the meter should be set correct at zero according to the maker's directions.

(b) Sensitiveness of recording meters may be examined by quickly stopping the flow of steam, and noting the time and character of the curve drawn when the pen returns to zero. Another useful test is to note the minimum amount of change in the valve opening regulating the flow, to produce a change in the recorder pen position. From this can be found the smallest change in the flow rate to which the meter will respond.

A Simple Orifice Meter. When steam or gases pass through an orifice, dropping the absolute pressure from P to p, the flow increases with increased pressure drop until the pressure in the throat of the orifice or nozzle reaches a certain value known as the *critical pressure, p'*. The usual method of expressing this condition is by means of the *critical pressure ratio, $r_c = p/P$*. The values usually given for r_c are 0.58 for saturated steam, 0.55 for superheated steam and 0.53 for air.

It should be noted that experiments have shown that *the critical pressure phenomenon does not hold for thin plate, sharp-edged orifices.*

The values of the critical pressure ratio given are substantially correct if the velocity of approach is negligible. Where approach velocity cannot be neglected, the computation of the value of r_c becomes quite laborious. It is usually much easier to determine the value by experiment.

A convenient method of orifice measurement, when r_c is less than the critical value, depends upon the application of Napier's formula, namely,

$$W = \frac{PA}{70}$$

in which W is the weight in pounds per second discharged, P is the absolute pressure, pounds per square inch, on the orifice, and A is the area of the orifice in square inches. Unless the entrance of the orifice is well-rounded, this expression gives results which are on the order of 5 per cent too high.

Napier's formula is based upon saturated steam but, should the steam be initially superheated, the weight discharged as given by the formula should be multiplied by the correction factor:

$$f = \frac{1}{1 + 0.00065T_s}$$

where T_s is the number of degrees of superheat in degrees F.

Emswiler gives a correction factor to be used when the steam is initially wet:

$$f = \frac{1}{1 - 0.012m}$$

where m is the percentage of moisture in the steam. It will be noted that, for small amounts of moisture in the steam, this correction factor can be neglected without causing serious error.

For cases above the critical pressure ratio, where the Napier formula does not apply, the following equation may be used:

$$W = \frac{Ap}{42} \sqrt{\frac{3(P - p)}{2p}}$$

where W is the weight discharged in pounds per second, A is the area of the orifice in square inches, P is the initial pressure in pounds per square inch, and p is the final pressure in the same units.

PART TWO

THE ANALYSIS OF COMBUSTION

THE CONSTITUENTS OF FUELS

The principal commercial fuels are coals, oils, and gases in their various forms. The elemental constituents of these fuels, which for the most part are common to all of them, are carbon, hydrogen, oxygen, nitrogen, and sulfur. Of these constituents the ones depended upon to yield heat through combustion are carbon and hydrogen. Sulfur has a heat value, but it is an undesirable element in fuel since in coal it goes to form clinker and in gas it may combine with water to make sulfuric acid, these processes accompanying the combustion of the fuel. Nitrogen and carbon in the form of CO_2 are inert gases and are valueless to combustion.

Carbon and hydrogen occur free or in combination with each other as hydrocarbons or in combination with oxygen. The only combustible carbon-oxygen compound is carbon monoxide, which is one of the most important of fuel gas constituents.

Coals are classified broadly as anthracitic, bituminous, sub-bituminous and lignitic. The anthracitic class contains three groups: meta-anthracite, anthracite and semi-anthracite which are distinguishable by the amounts of *fixed carbon* and *volatile matter* they contain. This is burnable material, mainly hydrocarbons so far as the volatile matter is concerned. The second class contains five groups: low volatile, medium volatile and three high volatile groups, *A*, *B*, and *C*. The last three have less than 69 per cent fixed carbon, and more than 31 per cent volatile (on the moisture and ash-free basis) but are distinguished by their B.t.u. content on the basis of the natural moisture which they contain when mined. The sub-bituminous class has three groups, *A*, *B*, and *C*, which are differentiated by heating value on the basis of natural moisture. The last group, lignitic coals, have two groups: lignite and brown coal which are classified on a B.t.u. basis. These classifications are contained in the A.S.T.M. Standards on Coal and Coke, published by the American So-

ciety for Testing Materials, Table I, p. 80, entitled "Classification of Coals by Rank."

The simplest form of coal analysis is called the proximate analysis which reports the percentages of moisture, fixed carbon, volatile matter and ash in the coal. The volatile matter is a more or less arbitrary definition of that part of the make-up of the coal, mainly hydrocarbons, which is driven off when the coal is heated under certain conditions. The moisture is driven off first by moderate heat, then the volatile matter. Finally, the coal is burned down completely to obtain the weight of the ash and the fixed carbon is obtained by difference.

A complete chemical analysis of coal, called the ultimate analysis, reports the make-up of the substance by chemical elements: carbon, hydrogen, oxygen, nitrogen and sulfur. Even the moisture is reported as part of the hydrogen and oxygen. When such an analysis is required it should be obtained from a chemical laboratory. The methods are far removed from mechanical experimentation and the apparatus required is usually not to be found outside of a well-equipped chemical laboratory.

The proximate analysis, however, may readily be made by the skilled engineer. Certain specific apparatus is required but is easily obtainable.

The kind of coal and its properties as fuel are determined roughly by the locality at which it is mined, but it should be noted that characteristics may vary markedly in samples even from the same mine.

The commercial grades of coal with relation to size are as follows, being listed in the order of the size, largest first.

Size, In.	Anthracite	Bituminous
4	Broken	Run of mine
3	Egg	Lump, various sizes
2	Stove	Nut, various sizes
$1\frac{1}{4}$	Chestnut	Screenings, various sizes
$\frac{7}{8}$	Pea	Washed sizes
$\frac{7}{16}$	Buckwheat, No. 1	
$\frac{1}{4}$	Buckwheat, No. 2	
$\frac{1}{8}$	Buckwheat, No. 3	

Oils used for fuel are composed of different hydrocarbons of the form C_nH_{2n+2} or C_nH_n, which have a wide range in such physical properties as specific gravity, volatility, burning and flash points. Gasoline and kerosene are the lighter distillates from crude oil.

The percentages by weight of carbon and hydrogen in various fuel oils and their distillates from all parts of the world are not very different.

Carbon is generally between 83 and 87 per cent and hydrogen between 11 and 15 per cent, the rest being oxygen, nitrogen and sulfur.

Gases. The principal gases used for fuel are illuminating gas, producer gas, natural gas, and blast furnace gas. The following table gives the percentages by volume of the constituents of representative American gases.

CONSTITUENTS OF GAS FUELS FOR POWER

("Low" Means Less than 1 Per Cent)

	Producer	Illuminating	Natural
Carbon monoxide, CO..........	25	15	low
Hydrogen, H_2...............	12	45	2
Methane, CH_4...............	2	25	95
Ethylene, C_2H_4 Benzol, C_6H_6 .... Heavy hydrocarbons	low	5	low
Oxygen, O_2..................	low	1	low
Sulfurated hydrogen, H_2S Sulfur dioxide, SO_2	low	low	low
Carbon dioxide, CO_2...........	5	4	low
Nitrogen, N_2.................	55	5	2

Blast furnace gas is combustible mainly through CO. It contains a little hydrogen and methane, and is high in nitrogen.

Analysis of these gases must be made by a chemist, with the exception, possibly, of producer gas and blast furnace gas. These may be analyzed by the Orsat apparatus if a pipette for the determination of hydrogen is provided, methane being ignored.

39. PROXIMATE ANALYSIS OF COAL

(a) Sampling. When an analysis of a small sample is made to represent a large lot of coal, every precaution should be taken to secure a truly representative sample. The probable error in sampling varies inversely with the weight of sample taken. Where a high degree of accuracy is required, the sample should be taken by one of the standard methods such as the Specification D 21-40 of the A.S.T.M. Standards. This method is also specified in the A.S.M.E. Code on Solid Fuels but, at the time of writing, this code is undergoing a revision which, if adopted,

will somewhat simplify the method of taking the sample. Since this new method has not been officially adopted it is best omitted at this time.

In cases where less accurate results are permissible, the following method may be used in collecting the sample. As each barrowful or charge of coal is removed from the pile for firing, an amount varying between a handful and a shovelful is withdrawn and put in a closed receptacle to make up a gross sample. The quantity of coal in each of these samplings and of the gross sample necessary for accurate results depends upon the lump size of the coal, upon its homogeneity, and to a limited extent upon the total amount used. For the smallest sizes of coal of fairly uniform quality, the gross sample may be no more than 100 lb.; under other conditions, it need not exceed 250 lb. Assuming 200 lb., the gross sample is broken on a clean surface, preferably a piece of sheet iron, with the assistance of a tamping bar or weight, so that the largest lumps are not more than ½ in. in diameter. This coal is then mixed and piled in the form of a cone. The cone should be "quartered" into four heaps by passing a board through the cone axis in two planes at right angles. Two diagonally opposite heaps are then discarded and the remaining two mixed, broken to ¼ in., coned and again quartered. The process of quartering and discarding should be repeated until about 5 to 10 lb. of coal lumps, not larger than ³⁄₁₆ in., remain. This sample should be placed in an air-tight container and sent to the laboratory. It is frequently advisable to divide the sample into two parts; one to be sent to the laboratory and the other to be retained until a satisfactory analysis of the laboratory has been completed.

(b) Preparation of the Laboratory Sample. The following method of preparing the laboratory sample is quoted from the A.S.T.M. Specification D 271-44. The same specification is used in the A.S.M.E. Code on Solid Fuels.

Coal Appearing Dry. (a) The sample which has been collected and reduced in accordance with the Standard Method of Sampling Coal for Analysis (A.S.T.M. Designation D 21) shall be crushed to pass an 840-micron (No. 20) sieve by passing through rolls or an enclosed grinder and a 50-g. total moisture sample taken, without sieving, immediately after the material has passed through the crushing apparatus. This sample should be taken with a spoon from various parts of the product passing an 840-micron (No. 20) sieve, and should be placed directly in a rubber-stoppered bottle.

(b) Thoroughly mix the main portion of the sample, reduce on the small riffle sampler to about 200 g., and pulverize to pass a 250-micron (No. 60) sieve by any suitable apparatus without regard to loss of moisture. After all the material has been passed through the 250-micron sieve, mix and divide it on the small riffle sampler to about 50 g. Transfer the final sample to a 4-oz. rubber-stoppered bottle. Determine the

THE PREPARATION OF A 1,000-POUND SAMPLE

NECESSARY TOOLS: SHOVEL, TAMPER, BLANKET (MEASURING ABOUT 6 BY 8 FEET), BROOM, AND RAKE. USE RAKE FOR RAKING OVER COAL WHEN CRUSHING IT, SO THAT ALL LUMPS WILL BE CRUSHED. SWEEP FLOOR OR BLANKET CLEAN OF ALL DISCARDED COAL AFTER EACH TIME SAMPLE IS HALVED OR QUARTERED.

FIRST STAGE

NOTE

Select a hard, clean surface, free of cracks and protected from rain, snow, wind, and beating sun. Do not let cinders, sand, chippings from floor, or any other foreign matter get into the sample. Protect sample from loss or gain in moisture.
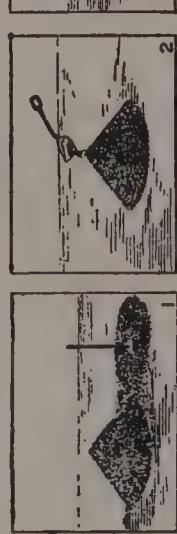
1. Crush 1,000-pound sample on hard, clean surface to 1-inch size.

2. 1,000-pound sample crushed to 1 inch and coned.

3. Mix by forming long pile. A—Spreading out first shovelful. B—Long pile completed.
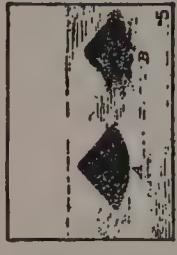
4. Halving by alternate shovel method. Shovelfuls 1, 3, 5, etc., reserved as 5, A; 2, 4, 6, etc., rejected as 5, B.
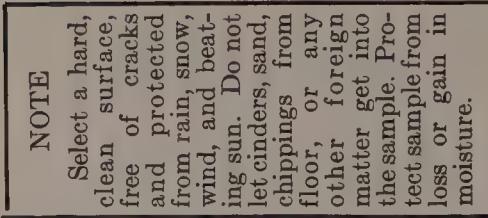
5. Long pile divided into two parts. A—reserve; B—reject.

SECOND STAGE

6. Crush 500-pound sample (Fig. 5, A) to ½-inch size.

7. 500 pounds crushed to ½ inch and coned.

8. Mix by forming long pile. A—Spreading out first shovelful. B—Long pile completed.

9. Halving by alternate shovel method. Shovelfuls 1, 3, 5, etc., reserved as 10, A; 2, 4, 6, etc., rejected as 10, B.

10. Long pile divided into two parts. A—Reserve; B—Reject.

THIRD STAGE

11. Crush 250-pound sample (Fig. 10, A) ½ to ¼-inch size.

12. 250 pounds crushed to ¼ inch and coned.

13. Mix by forming new cone.

14. Quarter after flattening cone.

15. Sample divided into quarters.

16. Retain opposite quarters A, A. Reject quarters B, B.

FOURTH STAGE

17. Crush 125-pound sample (Fig. 16: *A*, *A*) on blanket to pass a ½-inch size.

18. Mix by rolling on blanket.

19. Form cone after mixing.

20. Quarter after flattening cone.

21. Sample divided into quarters.

22. Retain opposite quarters *A*, *A*. Reject quarters *B*, *B*.

FIFTH STAGE

23. Crush 60-pound sample (Fig. 22: *A*, *A*) to ¼-inch size.

24. Mix by rolling on blanket.

25. Form cone after mixing.

26. Quarter after flattening cone.

27. Sample divided into quarters.

28. Retain opposite quarters *A*, *A*. Reject quarters *B*, *B*.

SIXTH STAGE

29. Crush 30-pound sample (Fig. 28: *A*, *A*) to $\frac{3}{16}$ inch or to pass a 4760-micron (No. 4) sieve.

30. Mix by rolling on blanket.

31. Form cone after mixing.

32. Quarter after flattening cone.

33. Sample divided into quarters.

34. The laboratory sample to be taken from *A*, *A*.

Fig. 88.—Standard Methods of Sampling Coal.

moisture in both the 250-micron sieve sample and the 840-micron sieve sample in accordance with Sections 7, 8, and 9 under the Determination of Moisture (in this same Specification).

(c) *Calculation.* Calculate the analysis of the coal passing the 250-micron sieve, which has become partly air-dried during sampling, to the dry-coal basis by dividing each result by 1 minus its content of moisture. Compute the analysis of the coal "as received" from the dry-coal analysis by multiplying by 1 minus the total moisture found in the sample passing an 840-micron sieve.

Coal Appearing Wet. (a) Spread the sample on tared pans, weigh, and air-dry at room temperature, or in a special drying oven at 10 to 15° C. above room temperature, and weigh again. Continue the drying until the loss in weight is not more than 0.1 per cent per hr. Drying should not be continued beyond this point because of the oxidation of the coal. Complete the sampling as described for dry coal.

(b) *Calculation.* Correct the moisture found in the air-dried sample passing the 840-micron sieve to total moisture "as received," as follows:

$$M = \frac{100 - A}{100} \times S + A$$

where M = total moisture of coal "as received";
A = percentage of air-drying loss;
S = percentage of moisture in air-dried sample passing an 840-micron sieve.

(c) Calculate the analysis to dry-coal and "as received" bases as described for dry coal, using for the "as received" calculation the total moisture as found by the formula in paragraph (b) in place of the moisture found in the coal passing an 840-micron sieve.

Freshly mined or wet coal loses moisture rapidly on exposure to the air of the laboratory, hence the sampling operations between opening the container and taking the total-moisture sample passing a No. 20 sieve must be conducted with the utmost dispatch and with minimum exposure to air.

The accuracy of the method of preparing laboratory samples should be checked frequently by resampling the rejected portions and preparing a duplicate sample. The ash in the two samples should not differ more than the following limits:

No carbonates present .... 0.4 per cent
Considerable carbonate and pyrite present................. 0.7 per cent
Coals with more than 12 per cent ash, containing consider-
    able carbonate and pyrite............................. 1.0 per cent

(c) Determination of Moisture.

The apparatus required for the determination of moisture in coal consists of an oven and a number of small porcelain crucibles. The oven should be constructed so that it will have a uniform temperature throughout its interior and the amount of air space should be kept as small as possible. There should also be provided a means of changing the air in the oven two or three times a minute. The entering air should be dried by causing it to pass through concentrated sulfuric acid before it enters the oven.

The porcelain crucibles should be of the low form, about ⅞ in. deep by 1¾ in. in diameter, and should be provided with a close-fitting flat aluminum cover. These same crucibles are suitable for the determination of fixed carbon and ash. Crucibles of fused silica may be used in place of the porcelain crucibles.

The empty crucibles should be heated under the same conditions at which the coal is to be dried. When thoroughly heated, they should be removed from the oven, covered, and placed in a desiccating dish filled with concentrated sulfuric acid, and allowed to cool for 30 min. When transferring to the balance, for weighing, the crucibles should be handled with tongs, otherwise they will pick up moisture if touched by the bare hand.

Two methods of weighing out the coal are in use. The *first* method consists of dipping out approximately 1 g. of coal from the sample bottle and quickly transferring this to the crucible. The cover should be put on immediately and the sample weighed at once. The *second* method consists of transferring slightly more than 1 g. of coal to the crucible and then bringing the weight to exactly 1 g. ± 0.5 mg. by quickly removing the excess weight of coal with a small spatula. This method is more open to error than the first and requires more skill, but, once mastered, it will be found more convenient.

The crucibles, containing the weighed samples, are next uncovered and quickly placed in the drying oven where they are allowed to remain for 1 hr. at a temperature between 104 and 110° C. Upon being removed from the oven, the crucibles should be covered at once and placed in a desiccating dish to cool. When cool, the crucibles are again weighed, the loss of weight being taken as the moisture, which is reported in per cent.

The above method refers to the 60-mesh sample. The 20-mesh sample may be treated in the same way except that 5 g. are taken and the heating is continued for 1½ hr.

(d) Determination of Volatile Matter. This portion of the proximate analysis requires strict attention to detail. The sample of coal is weighed in a tared platinum or illium crucible, of 20 to 30 cc. capacity. The crucible must be provided with a tightly fitting cover. The crucible is then placed in an electric tube furnace or an electrically or gas-heated muffle furnace. The cover of the crucible should be placed lightly at first. The temperature of the furnace must be maintained at 950° C. (±20°). When the rapid evolution of volatile matter has ceased, as evidenced by the disappearance of the luminous flame, the cover should

be tapped to make a tight seal against the admission of air and the heating then continued for exactly 7 min. At the end of that time the crucible should be removed from the furnace and, when cool, weighed. The loss of weight minus the moisture is the volatile matter.

Note: The later editions of the A.S.T.M. Standards require the use of platinum crucibles for the volatile matter and fixed-carbon determinations. This equipment is probably too expensive for every-day student use; the porcelain crucibles will serve for this purpose.

In certain cases a modification of the above method must be employed. Sub-bituminous coals, lignites and peat suffer mechanical losses when heated suddenly. To prevent this, the crucible should be given a preliminary heating by playing the flame of a Bunsen burner over the crucible for about 5 min. This should be done in such a manner that the discharge of the volatile matter will not cause sparking. The crucible is then transferred to the furnace and heated for 6 min. at 950° C.

(e) Determination of Ash. The porcelain crucibles, containing the dried sample from the moisture determination, are placed in a cold muffle furnace and gradually heated to redness at such a rate as to prevent any mechanical losses due to the rapid discharge of volatile matter. After the volatile has been discharged, the ignition is finished at a temperature between 700 and 750° C. until the weight becomes constant (± 1 mg.). The crucible is then removed from the surface, cooled in a desiccator, and weighed when cool.

(f) Determination of Fixed Carbon. The fixed carbon is found by difference, as follows:

$$100 - (\text{moisture} + \text{ash} + \text{volatile matter}) = \text{fixed carbon},$$

all values being expressed in per cent.

(g) Determination of Sulfur. Quite frequently it is necessary to know the amount of sulfur in coal without requiring an ultimate analysis. Sulfur may be determined separately and is frequently reported with a proximate analysis. The method used is that of Eschka and a complete description will be found in the A.S.M.E. Test Code for Solid Fuels.

(h) Permissible Differences in Duplicate Determinations. The following table gives the permissible differences in duplicate determinations by the same or a different analyst.

(i) Calculation of Hydrogen and Volatile and Total Carbon from the Proximate Analysis. For the complete analysis of boiler and furnace

performance, it is necessary to know the percentage of total carbon in the fuel and of hydrogen in the volatile matter. As the proximate analysis shows only the fixed carbon, that in the volatile matter may be estimated and added to the amount of fixed carbon to get the total. Professor Lionel S. Marks provides an empirical method of doing this and of estimating the hydrogen in the volatile matter. He has pointed out a relation applying to American coals by means of curves, and these

TABLE OF PERMISSIBLE DIFFERENCES

Determination	Permissible Difference	
	Same Laboratory, Per Cent	Different Laboratory, Per Cent
Moisture (less than 5 per cent)..............	0.2	0.3
Moisture (over 5 per cent)................	0.3	0.5
Volatile (bituminous coals)................	0.5	1.0
Volatile (lignites)........................	1.0	2.0
Ash (no carbonates present)...............	0.2	0.3
Ash (carbonates present)..................	0.3	0.5
Ash (coals with more than 12 per cent ash and containing carbonates and pyrites)...	0.5	1.0
Sulfur (under 2 per cent).................	0.05	0.1
Sulfur (over 2 per cent)..................	0.1	0.2

curves have been expressed, in part, by the following equations of Professor Diederichs:

Let h_c = hydrogen, exclusive of that in moisture;
c_c = carbon in the volatile matter or "volatile carbon";
v_c = volatile matter:

all expressed as weight-percentages of the combustible. Then

$$h_c = v_c \left(\frac{7.35}{v_c + 10} - 0.013 \right) ;$$

$$c_c = 0.02 v_c^2 \text{ or } 0.9(v_c - 10) \text{ for anthracites;}$$

$$c_c = 0.9(v_c - 14) \text{ for bituminous coals.}$$

As an example of the use of these formulas, take the proximate analysis given and let it be required to find the hydrogen in 1 lb. of the coal, H_t, and the total carbon, C_t.

The combustible, being the sum of the fixed carbon and volatile matter, is $0.807 + 0.0617 = 0.869$ lb. per lb. of coal. Hence

$$v_c = \frac{0.0617}{0.869} \times 100 = 7.1\%,$$

and

$$h_c = 7.1 \left(\frac{7.35}{7.1 + 10} - 0.013 \right) = 3 \text{ per cent.}$$

This is the percentage of *hydrogen based on the combustible.* To base it on the coal, we have

$$H_t = \tfrac{3}{100} \times 0.869 = 0.026 \text{ lb. of hydrogen per lb. of coal.}$$

Similarly for the volatile carbon,

$$c_c = 0.02 \times 7.1^2 = 1 \text{ per cent.}$$

Consequently in 1 lb. of coal, there is $\tfrac{1}{100} \times 0.869 = 0.00869$ lb. of volatile carbon, and the total carbon is

$$C_t = 0.807 + 0.00869 = 0.816 \text{ lb.}$$

40. Combustible in Ash and Refuse

(a) **Sampling.** The sampling of ash and refuse in boiler tests should be made in the same manner as that described for sampling coal. During the process of reduction a grab sample may be taken for moisture, this sample to be transported to the laboratory in a sealed container. The laboratory sample, for combustible, should also be sent to the laboratory in a sealed container.

(b) **Treatment of Moisture Sample.** On arrival at the laboratory the moisture sample should be quickly crushed, in a jaw crusher, to pass a 4-mesh sieve and reduced to about 5 lb. in a riffle. The sample is then to be weighed immediately and spread on galvanized-iron pans which are placed in an air-drying oven at a temperature not over 200° F. The samples are dried in the oven until the weigh loss becomes not more than 0.1 per cent per hr.

The sample for combustible should be crushed, mixed, quartered and reduced until a small sample remains which will pass a 60-mesh sieve.

This may be dried in a manner similar to coal samples and finally treated in the manner described for the determination of ash in coal. The difference of weight before and after ignition is the combustible.

THE HEAT VALUE OF FUELS

When used for the generation of power, combustion may be defined as the rapid chemical combination of the oxygen in air with the combustible constituents of fuel, which combination is accompanied by the evolution of heat. It has been pointed out that the elemental combustibles in fuels are carbon, hydrogen, and, to a lesser degree, sulfur. The complete combustion of carbon forms carbon dioxide, CO_2, and hydrogen, water, H_2O.

When a unit mass of fuel is burned completely, the heat evolved raises the temperature of the materials entering into the combination and of surrounding objects. If the products of combustion are cooled to the temperature before combustion, then the total heat given up is the "heat" or "calorific value" of the fuel. Or, more briefly, *the heat value of a fuel is the number of heat units that are released by the complete combustion of a unit mass of the fuel.*

It should be noted that to release all of the heat generated, the products must be cooled down to room temperature. This may be done in two ways, namely, so that the H_2O formed by combustion of the hydrogen remains as steam or so that the H_2O is condensed. In the former case the latent heat of the steam remains in the products of combustion, the heat released is correspondingly less, and is referred to as the "lower heat value." In the latter case the latent heat of the steam is included in the heat released which is then called the "higher heat value."

Whether the one or the other quantity should be used depends upon the character of the test for which the heat value is needed.

The difference between the two values depends upon the amount of hydrogen in the fuel. For coals, it is comparatively small, but for gases and oils it may be as high as 15 per cent.

The units in which heat values are expressed in the United States are as follows, corresponding metric units being used to a limited extent only.

For solids, B.t.u. per pound.
For liquids, B.t.u. per pound.
For gases, B.t.u. per standard cubic foot.

The standard cubic foot of gas is a cubic foot of gas under standard conditions of pressure and temperature. That this specification is necessary will be seen when it is considered that the mass of gas in a cubic foot depends upon these conditions, and consequently its heat value. The standard conditions of pressure and temperature are either 29.92 in. of mercury and 32° F., or 30 in. of mercury and 68° F. These pressures are the same and equal to 14.7 lb. per sq. in., the difference in the mercury columns being due to the temperature differences. The American Gas Association uses 60° F. as the standard reference temperature. In this work the 32° standard will be used.

The following gives experimentally determined heat values of the elemental combustibles and of various gases, and of fuels.

HEAT VALUES IN B.T.U. PER POUND

Carbon burned to CO_2.......... 14,520
Carbon burned to CO.......... 4,340
Sulfur burned to SO_2........... 3,982
Hydrogen to H_2O............. 61,045

HEAT VALUE AND CONSTANTS FOR GASES AND VAPORS
(Abstracted from A.S.M.E. Code on Gaseous Fuels)

Gas or Vapor	Formula	Mol. Wt.	Heat Value, B.t.u./lb. (higher)	Heat Value, B.t.u./cu. ft. (Higher)	Heat Value, B.t.u./cu. ft. (lower)
Hydrogen.............	H_2	2.016	60,983	318.9	269.6
Carbon monoxide......	CO	28.010	4,343	315.8	315.8
Methane.............	CH_4	16.042	23,873	995.8	897.0
Ethane...............	C_2H_6	30.069	22,314	1,755	1,606
Propane.............	C_3H_8	44.095	21,654	2,521	2,320
Acetylene............	C_2H_2	26.036	21,490	1,462	1,413
Ethylene.............	C_2H_4	28.052	21,635	1,584	1,485
Propylene............	C_3H_6	42.079	21,033	2,333	2,183
Butylene.............	C_4H_8	56.105	20,829	....	....
Amylene.............	C_5H_{10}	70.131	20,693	....	....
Hydrogen sulfide......	H_2S	34.076	7,097	633	583
n-Butane.............	C_4H_{10}	58.121	21,301	3,324	3,068
iso-Butane...........	C_4H_{10}	58.121	21,250	3,296	3,043
n-Pentane...........	C_5H_{12}	72.147	21,084	3,948	3,652

Note: Values are for temperatures of 68° F. and 29.92 in. of Hg. at 32° F.

Values originally supplied by R. S. Jessup of the National Bureau of Standards, Washington, D. C.

APPROXIMATE HIGHER HEAT VALUES OF COMMERCIAL FUELS, B.T.U.

Coals..........................	11,000 to 15,000 (per lb.)
Oils...........................	18,000 to 20,000 (per lb.)
City gas, average.................	550 (per cu. ft.)
Producer gas, average.............	150 (per cu. ft.)
Natural gas, average..............	1,050 (per cu. ft.)

If it is desired to refer the heat values of the gases to the 32° F. standard, it is only necessary to multiply by 1.073, the ratio of absolute temperatures.

Instruments for determining the heat value of fuels are called "fuel calorimeters."

41. The Determination of the Heat Value of Coal

Principles. This may be done by calculation from empirical formulas involving the results of the proximate or ultimate analysis of the coal, or by the use of a fuel calorimeter. The latter method is the more accurate.

Results from coal calorimeters are obtained by burning a small sample of the coal in an air-tight chamber immersed in water. The heat given up to the water and calorimeter parts by the combustion of this coal is measured by the rise of temperature, from which the heat value of the coal may be calculated.

The essential parts of a coal calorimeter are the water vessel, jacketed to prevent radiation, the combustion chamber or "bomb" with a device for igniting the charge, and a thermometer or its equivalent. Ignition is accomplished by an electric current which heats an iron or platinum wire to incandescence. To support combustion, oxygen is charged with the coal in the free state from a cylinder under pressure.

The **Atwater, Davis, Emerson, Mahler, Parr, Peters and Williams oxygen bomb calorimeters** are the only ones recognized as standard by the A.S.M.E., for solid fuels, and only these will be discussed in this work. In order to meet the standards set up by the Test Code for Solid Fuels (A.S.M.E.), only calorimeter bombs which are noncorrosive or have a noncorrosive lining of platinum, gold, porcelain or other metal not attacked by nitric and sulfuric acids or other products of combustion, may be used.

As a protection against air currents, the calorimeter *must* be provided with a water-jacket and cover. The water in the jacket must be kept

within 2 or 3° C., of room temperature and should be stirred continuously with some type of mechanical stirring device.

The water in the calorimeter can must also be stirred in order to give consistent thermometer readings while the temperature is rising rapidly. A motor-driven screw or turbine stirrer is suitable provided that the rate of stirring can be kept constant and also that the rate is not excessive. This latter may be determined by running the stirrer for 10 min. If the temperature rise is more than 0.01° C. in this time, the stirrer is running too fast. It is impossible to obtain accurate determinations if the stirrer is run too fast or the rate of stirring is not constant.

FIG. 89.—Emerson Fuel Calorimeter with Adiabatic Jacket.

The **thermometers** used should be graduated to 0.01° C. and should be tested by a government testing bureau and the corrections given on the certificate should be used. Corrections for emergent stem should be made in every case. In addition, the stem of the thermometer should be tapped lightly before reading in order to avoid errors caused by sticking of the meniscus. Beckman thermometers are suitable but less convenient than the regular calorimetric thermometer.

(a) **Preparation of the Sample.** Sampling of the coal should be done exactly as for the proximate analysis. For bituminous coals a fineness of 60-mesh is sufficient but for anthracite coals and coke it is better to grind somewhat finer. The ground coal must be thoroughly mixed in the bottle before taking out a sample for testing purposes.

(b) **Determination of Heat Value by the Emerson or Mahler Calorimeter.** Let W represent the weight of calorimeter water in grams, and w the water equivalent of the bomb, water vessel and other parts in contact with the calorimeter water (that is, the weight of water that would absorb the same amount of heat as these parts); and T the temperature, degrees C., rise brought about by the combustion of C grams of coal. The heat given up by the combustion is then $(W + w) \times T$ calories, and the heat value of the coal is $(W + w)T \div C$ calories, per gram. To convert this into B.t.u. per pound, multiply by 1.8. The temperature rise,
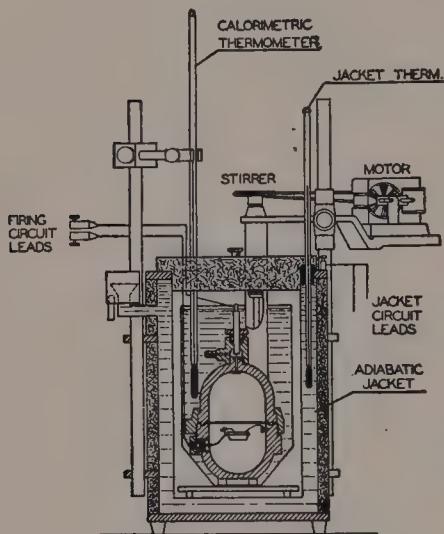
T, as observed, must be corrected as will be described. More detailed instructions for operation follow:

A small test tube containing about 2 g. of the pulverized coal is weighed on a chemical balance to an accuracy of 1 mg. About 1 g. of this coal is poured into the pan provided for the bomb charge (not less than 0.8 g. or more than 1.2 g.). The test tube with the remaining coal is then weighed again, and the weight of the bomb charge obtained by difference.

For anthracite, coke, and coal of high ash content, which do not readily burn completely, the following procedure is recommended:

The inside of the fuel pan is lined completely with ignited asbestos in a thin layer pressed well down into the angles. The coal is then sprinkled evenly over the surface of the asbestos. Otherwise the procedure is as previously described.

The firing wire, if iron, is measured and coiled into a small spiral and connected between the terminals. Approximately 0.5 cc. of water is placed in the bottom of the bomb to saturate the oxygen used for combustion. When the fuel pan is placed in the bomb, the firing wire should touch the coal or other material to be tested.

The cap should next be screwed in place, a little vaseline on the threads enabling a tighter fit. Care must be taken that no vaseline be in the combustion space. Oxygen, from a supply tank, is then admitted slowly in order to avoid blowing the coal out of the fuel pan. The pressure should be allowed to reach 20 atmospheres for the large bombs and 30 atmospheres for the smaller bombs. This practice insures sufficient oxygen for complete combustion.

The calorimeter is filled with the required amount of distilled water, which may either be weighed or measured in a standardized flask. In every case, the amount must be the same as that used for standardizing the calorimeter. The temperature of the water should be adjusted so that the final temperature will not be more than 1° C. above that of the jacket.

The bomb being ready for firing, the firing wire is attached and the bomb placed in the calorimeter. The cover is then put in place and the stirrer and thermometer placed so as not to be in contact with the bomb or container. The stirrer is started and after the thermometer reading has become steady, not less than 2 min. after starting the stirrer, temperatures are read every minute for a period of 5 min. and the charge fired at the end of that time. The firing switch should remain closed for not more than 2 sec.

After firing the charge, the temperature should be read every half minute, tapping the stem of the thermometer lightly a few seconds before the reading is taken. After the maximum temperature has been reached and is falling uniformly a second series of 1-min. readings are taken for 5 min. to determine the final rate of cooling.

The next step is to remove the bomb from the calorimeter and allow the gases to escape by opening the valve. When the pressure in the bomb is reduced to atmospheric pressure, the bomb is opened and the inside examined for traces of incomplete combustion. If any unburned material or sooty deposits are found the test must be discarded. If the combustion appears to have been complete, the bomb is rinsed with clean water and the washings titrated with a standard alkali solution using methyl orange or methyl red as an indicator. It is suggested that the standard solution be so proportioned that 1 cc. = 0.02173 g. of nitric acid = 5 calories correction for the heat of formation of nitric acid. In addition, a correction of 1300 calories per gram of sulfur in the coal should be made to account for the excess of difference in heats of formation of SO_2 and aqueous H_2SO_4 over the heat of formation of aqueous HNO_3.

(c) Computation of Results. The following method of computing results is recommended, by the A.S.M.E. Test Code for Solid Fuels, to take the place of the *Pfaundler* or other similar methods for computing the cooling or radiation correction.

Observe (1) the rate of rise (r_1) of the calorimeter temperature in degrees per minute for 5 min. before firing; (2) the time (a) at which the last temperature reading is made immediately before firing; (3) the time (b) when the rise of temperature has reached six-tenths of its total amount (this point can generally be determined by adding to the temperature observed before firing, 60 per cent of the expected * temperature

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* When the temperature rise is not approximately known beforehand, it is only necessary to take thermometer readings at 40, 50, 60 sec. (and possibly 70 sec. with some calorimeters) after firing, and from these observations to find when the temperature rise had reached 60 per cent of the total. Thus, if the temperature at firing was 2.135°, at 40 sec. 3.05°, at 50 sec. 3.92°, at 60 sec. 4.16°, and the final temperature was 4.200°, the total rise was 2.07°; 60 per cent of it was 1.24°. The temperature to be observed was then 2.14° + 1.24° = 3.38°. Referring to the observations at 40 and 50 sec., the temperatures were respectively 3.05 and 3.92°. The time corresponding to the temperature of 3.38° was therefore

$$40 + \frac{3.38 - 3.05}{3.92 - 3.05} \times 10 = 44 \text{ sec.}$$

rise, and noting the time when this point is reached); (4) the time (c) of a thermometer reading taken when the temperature change has become uniform some 5 min. after firing; (5) the final rate of cooling (r_2) in degrees per minute for 5 min.

The rate r_1 is to be multiplied by the time $b - a$ in minutes and tenths of a minute, and this product added (subtracted if the temperature was *falling* at the time a) to the thermometer reading taken at the time a. The rate r_2 is to be multiplied by the time $c - b$ and this product added (subtracted if the temperature was *rising* at the time c and later) to the thermometer reading taken at the time c. The difference of the two thermometer readings thus corrected, provided the corrections from the certificate have already been applied, gives the total rise of temperature due to the combustion. This multiplied by the water equivalent of the calorimeter gives the total amount of heat liberated. This result, corrected for the heats of formation of HNO_3 and H_2SO_4 observed and for the heat of combustion of the firing wire, when that is included, is to be divided by the weight of the charge to find the heat of combustion in calories per gram. Calories per gram multiplied by 1.8 give the B.t.u. per pound.

The permissible differences in duplicate determinations are as follows:

Same analyst.............. 0.3 per cent
Different analysts......... 0.5 per cent

In practice, the time $b - a$ will be found so nearly constant for a given calorimeter with the usual amounts of fuel that b need be determined only occasionally.

The results should be reduced to calories per gram or B.t.u. per pound of *dry coal*, the moisture being determined upon a sample taken from the bottle at about the same time as the combustion sample is taken.

The result obtained by the foregoing method of computation and determination is the total heat of combustion at constant volume, with the water in the products of combustion condensed to liquid at the temperature of the calorimeter, that is, about 20 to 35° C.

Net heat of combustion at 20° C. shall refer to results corrected for latent heat of vaporization, as follows:

Total heat of combustion in B.t.u. − 1040 (hydrogen × 9) = net heat of combustion in B.t.u. per pound. Also

Total heat of combustion in calories − 580 (hydrogen × 9) = net heat of combustion in calories per gram.

The method of computing the "cooling correction" described in *Technical Paper No. 8,* Bureau of Mines, pages 28 to 32, may also be used.

42. Standardizing Oxygen Bomb Calorimeters

The water equivalent of an oxygen bomb calorimeter may be found by burning a known weight of a standardized fuel, of known calorific value. Samples of standardized fuels may be obtained from the Bureau of Standards in Washington. The materials most commonly used are chemically pure cane sugar, benzoic acid, and naphthaline. These materials are prepared by the Bureau specifically for use in standardizing calorimeters. The samples are received from the Bureau in a finely divided condition. When using naphthaline, the sample should be briquetted or fused into a solid mass.

To standardize a calorimeter with one of these materials, the bomb, fuel pan and fuse wires are prepared in the same manner as in the testing of a fuel. Iron fuse wire is the best to use when making a standardization. The iron wire should be wound in a narrow spiral at the point where it touches the combustible. The pan should be about three-quarters filled with a known weight of the standard material with the iron wire resting on the surface of the same. The iron wire should be about 0.004 in. in diameter. Wires larger than this require heavier fusing current and are not desirable.

After the substance in the fuel pan has been weighed, it should be transferred to the bomb at once and the bomb closed. This is particularly necessary when using naphthaline to prevent loss of weight by sublimation. In making the run the weight of the combustible is recorded and the weight of the fuse wire burned. When using cane sugar as a combustible, a few grains of naphthaline are sprinkled on the wire helix to act as an igniter. The weight of naphthaline used must be known. The weight of fuse wire burned is found by weighing the entire piece of wire and subtracting the weight of any unburned ends found after combustion.

The following corrections must be made:

1. The heat generated by the small quantity of naphthaline used as igniter.
2. The heat generated by the burning of the fuse wire.
3. The heat input of the electric current used in bringing the fuse wire to incandescence.
4. The heat of formation of nitric acid.

The heat of combustion of the naphthaline should be taken from the label on the standard sample. The heat of combustion of the iron wire is 1600 calories per gram. The correction for electrical input can best be determined by a blank run in which wire, of the same diameter as that used in the test, is burned in the bomb without a charge of combustible. The blank run is made with the temperature in the calorimeter can exactly the same as the surrounding conditions, in order to avoid a cooling correction. When the temperatures within and without the calorimeter are exactly equalized, the current is turned on for an exact period of time (not more than 2 sec.). The "current on" period should be exactly duplicated when the standardization run is being made with the standard combustible. From the total calories developed in the blank run should be subtracted the heat generated by the burning of a known weight of iron wire. The remainder is the heat generated by the flow of current.

The correction for the formation of nitric acid may be obtained by titration of the bomb washings as previously explained.

The corrections are all subtractive and are deducted from the results obtained from the calorimeter test. Since the weight of water used in the calorimeter is also known, the water equivalent of the bomb and calorimeter parts may be computed from the data obtained in the standardization test. Not less than five such tests should be made in order to obtain an accurate standardization.

43. Calculation of Heat Value from Analysis

If the combustibles in coal were in the elemental form, it would be easy to calculate the heating effect from each element according to its amount and thus determine the heating value of the coal. This, however, is not so; they are combined as hydrocarbons and otherwise. Now when combustion takes place some heat must be given up to the hydrocarbons to separate the carbon from the hydrogen so that they may recombine with oxygen. That is, some heat becomes latent through dissociation of the elements which lessens the heat available from their combination with oxygen. So there is an interchange of heat in burning coal which makes its heat value dependent upon the previous combination of its elements. We therefore cannot calculate heat values with precision in this way, but certain empirical formulas enable us to make fair estimates. One of them is Dulong's as below.

$$\text{Heat value} = 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4000S,$$

the symbols C, H, O, and S standing for the weights in pounds of carbon, hydrogen, oxygen, and sulfur in 1 lb. of the fuel, respectively. The result is in B.t.u. per lb. This formula uses the ultimate analysis of the coal, and is more applicable to anthracite than to bituminous coals.

Another formula, using the proximate analysis, is Goutal's. This is

$$\text{Heat value} = 147.6 \times fc + K \times vm,$$

in which fc and vm are the percentages of fixed carbon and volatile matter in the fuel as received, respectively, and K has different values depending upon the amount of volatile matter, as shown in the curve, Fig. 89a.
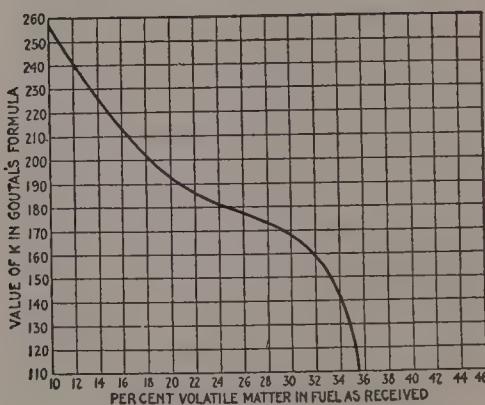
Fig. 89a.

Fig. 89b is a reproduction of Mahler's curve, from which, probably, may be had as reliable a result of heat value as any of the relations proposed
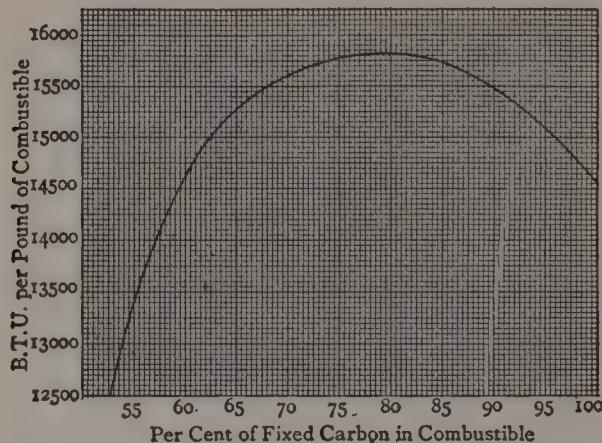
Fig. 89b.—Mahler's Curve for Coal Heat Values.

for this purpose. As an example of its use assume a coal with 16.9 per cent of volatile matter and 70.8 per cent of fixed carbon. Then

Combustible in the coal   = 16.9 + 70.8 = 87.7 per cent;

Fixed carbon in the combustible = 70.8 ÷ 87.7 = 80.6 per cent.

From Fig. 89b, against 80.6 per cent is found the heat value of the combustible, 15,800 B.t.u. per lb. Consequently, the

Heat value of the coal = 15,800 × 0.877 = 13,900 B.t.u.

It should be emphasized that all of these relations for calculating the heat value of coal are approximate, and generally give better values with the lower percentages of volatile matter, the probable error then being within 2 per cent.

44. The Determination of the Heat Value of Gases

Principles. The heat values of gases may be calculated approximately from their chemical analyses. Gases may be tested for heat value by the use of a properly designed calorimeter. The constant pressure, flow type calorimeter is universally used for this purpose.

This instrument is an ingenious arrangement of heating surfaces surrounded by water, by which all the heat generated by the fuel to be tested is passed to the water. The water is kept flowing through the calorimeter at a constant rate, secured by keeping it under a constant head, and enters continuously at a uniform temperature. Upon leaving the calorimeter, it may be weighed, and the heat added ascertained by noting its rise of temperature. In order that all the heat be given up to the water, the products of combustion must return to the temperature of the fuel and air from which they came, before leaving the calorimeter. This is ascertained by a thermometer placed in the exit gas flue. The water resulting from the burning of hydrogen is condensed and collected, should it be desired to calculate the lower heat value.

To burn the gas, a Bunsen burner is used and a small gas meter is employed to measure the fuel in cubic feet.

This meter is of the "wet meter" type, that is, it contains a water seal. Gas samples after passing through it are delivered to the calorimeter 100 per cent humid. The conversion of the metered volume of fuel into standard cubic feet must account for this humidity.

For liquid fuels, a special regenerative burner is used which, together with the lamp containing the fuel, is attached to one arm of a beam balance so that the weight of fuel burned may be measured at any time during the test.

There is no water equivalent of the calorimeter to be considered since, during operation, all parts are at constant temperatures.

Radiation is made negligible by air jacketing and polished surfaces.

A correction must be made for the humidity of the air used in combustion. This is best done by referring to the chart, page 9, of the A.S.M.E. Test Code on Gaseous Fuels (1944).

Sampling. If the sample to be tested represents a gas used in a test of several hours' duration, it is best to take a continuous sample covering the whole time of the test as described for exhaust gas sampling. This involves rather a large gas container, however, and it is more convenient to draw the gas from the main directly into the calorimeter. For this purpose the main should be tapped at a point as near as possible to the place where the gas is used. Then, if a number of heat value determinations are made covering equal intervals of time and separated by equal intervals, their average will be the average heat value of the total gas delivered in the main.

For tests requiring accuracy within 2 or 3 per cent of the heat value of the gas, the humidity of the gas in the main should be measured, especially if the gas is at a temperature higher than 80° F. A wet and dry bulb thermometer may be inserted in the gas stream, or a sample of gas may be drawn through a glass tube containing calcium chloride and then through a volume meter. The weight gained by the calcium chloride equals the weight of H_2O contained in the measured volume of gas.

(a) Determination of Higher Heat Value. The rates of water and gas flow should be adjusted so that the temperature of the products of combustion upon leaving the calorimeter is that of the room. Then, if t and T are the temperatures of the water before and after heating, respectively, in degrees F., W, its weight in pounds, and G, the cubic feet of gas burned, we may say

$$\text{Higher heat value} = (T - t)W \div G.$$

In this, G is the number of cubic feet of gas under standard conditions.

The American gas industry uses an arbitrary standard at 14.7 lb. per sq. in. and 60° F., the gas being 100 per cent humid. To convert the gas meter reading to cubic feet under these conditions, the following relation should be used,

$$G = G' \times \frac{520}{T} \times \frac{P - P_v}{29.5}$$
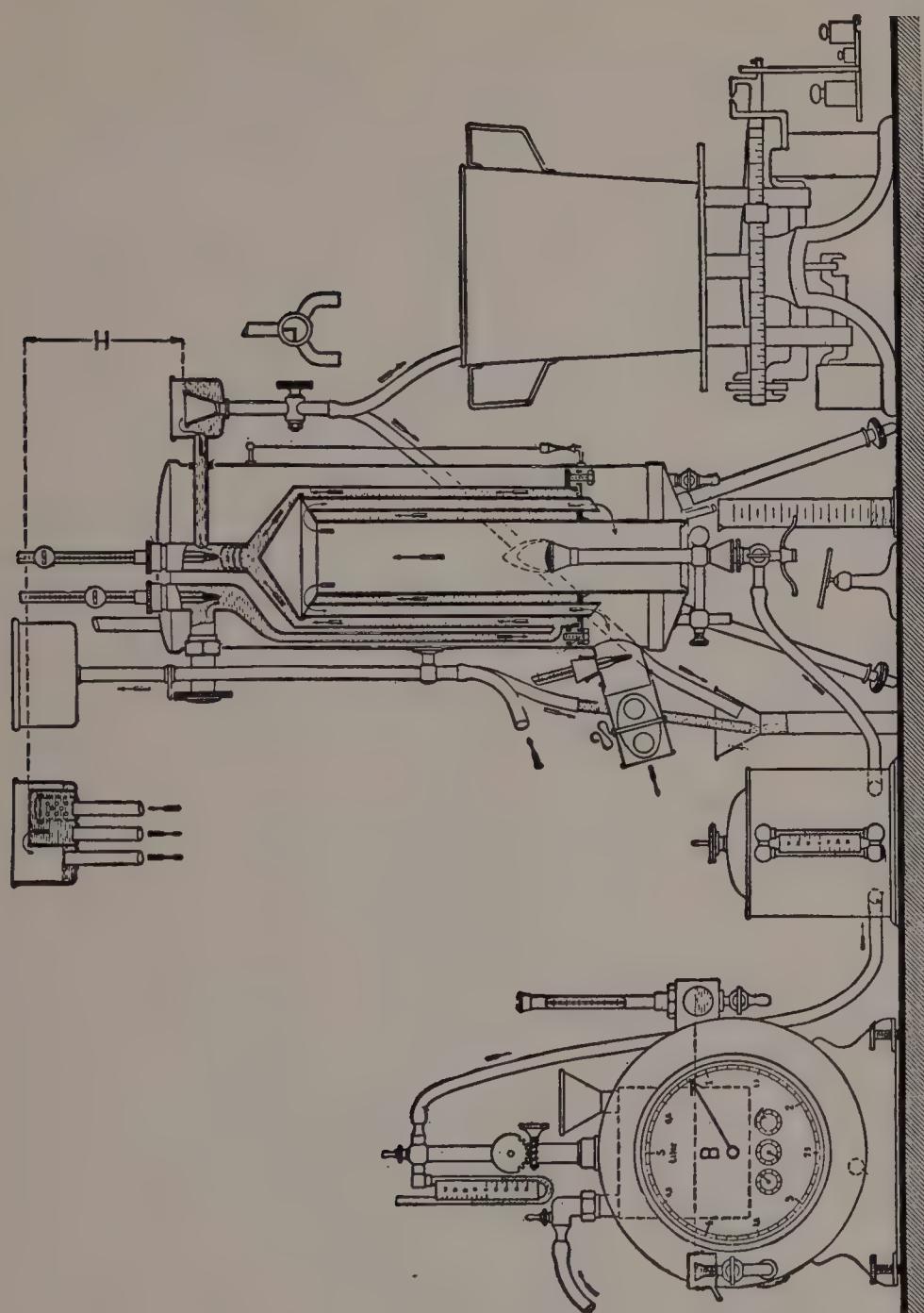
Fig. 90.—Junker's Calorimeter.

in which G' = the gas volume, by meter, in cubic feet;

T = the gas temperature at the meter, degrees F., absolute;

520 = the standard temperature;

P = the pressure of the gas at the meter in inches of mercury, equal to the barometric pressure plus the manometer pressure converted to inches of mercury;

P_v = the pressure of saturated steam in inches of mercury at the temperature, T;

29.5 = standard barometric pressure less that of saturated steam at 60°.

It will be noted that the correction for pressure is for the pressure of the dry gas, that is, for the total pressure less that of the partial pressure of 100 per cent humidity.

The conversion to standard cubic feet requires readings of barometer, manometer and thermometer in the gas main.

The heat value determined in this way is not quite the higher value, for the reason that not quite all the water of combustion is condensed. This is because the air entering combustion is not saturated with water vapor (although the fuel may be), but the products of combustion are saturated. The difference in the humidities is due to the water of combustion which is not condensed, the result being unaccounted for latent heat. A correction may be computed as described in the section on Total Heat of Air-Steam Mixtures in the Appendix. The determinations as made without this correction are acceptable for most purposes.

(b) Determination of Lower Heat Value. This is determined correctly by subtracting from the heat added to the water the amount which came from the vapor in condensing, or its latent heat. To determine this amount correctly, we should know the weight of the vapor and its latent heat per pound. The latent heat of steam is generally determined from its pressure by reference to the steam tables. In the case of steam mixed with other gases, the pressure of the steam is only partial and therefore difficult to determine. Under the conditions existing during the operation of the Junker calorimeter, it is convenient and sufficiently accurate to take the latent heat of the steam as that corresponding to the *temperature* of the exhaust gases, that is, room temperature. Letting L stand for this heat and w for the weight of condensation, we have

$$\text{Lower heat value} = \frac{(T - t)W - Lw}{G},$$

the other notation being as used under (a).

An average value of L may be taken as 1060 for temperatures between 50° and 70° F.

(c) Calculation of the Heat Value from the Fuel Analysis. This may be readily and acceptably done in the case of gases when the complete analysis is known. The following rule may be used:

To calculate the higher or lower heat value of a fuel gas, multiply the higher or lower heat value of each of the constituent gases, in B.t.u. per standard cubic foot, by its volume percentage, as shown by the fuel gas analysis, and divide by 100. Add these results together; the sum is the desired heat value.

This does not give exact results when the fuel contains hydrogen or hydrocarbons: first, because of the different H_2O content of the products when hydrogen only is burned with air from that when it is mixed with other gases; and, second, because the constituent analyzed as C_2H_4 is not only that compound, but represents other hydrocarbons, the heat value of which may not equal that of C_2H_4.

45. Heat Value of Oils by Oxygen-Bomb Calorimeter

The method of determining the heating value of liquid fuels by the Junker's calorimeter is suitable for light oils, gasoline and alcohol, but is not suitable for heavier fuel oils.

Heavy Oils. In the handling of liquid fuel the most suitable method of preventing evaporation of the samples is to hold the sample in a small weighing bottle with a dropper arranged in the cork of the bottle. The dropper is used to transfer the liquid from the bottle to the fuel pan. After introducing the oil into the fuel pan, the dropper and cork are replaced in the weighing bottle. The weight of the bottle with the dropper and cork is taken before and after the oil is removed; the difference in the two weights being the weight of oil in the fuel pan.

With heavy oils, the oil may be placed in the pan directly upon a small amount of ignited asbestos. As soon as the oil is placed on the asbestos, the bomb should be closed to prevent any evaporation of the sample. The remainder of the manipulation is exactly as that described for testing coal.

Light Oils. Light oils, gasoline, and alcohol may be tested in the oxygen-bomb calorimeter, but because of the rapid evaporation of these liquids it is not advisable to place them directly in the fuel pan. Small gelatine capsules, filled with ignited asbestos, are used into which the oil may be poured, the latter being absorbed by the asbestos. The filled

capsule is sealed, placed in the fuel pan and burned in the usual manner using iron fuse wire. The dry weight of the capsule and asbestos must be known and after filling the weight is again taken and the weight of oil found by difference. Care should be used that no air bubbles are enclosed with the charge in the capsule, as the fuel will otherwise ignite with explosive force spattering the oil on the walls of the bomb where it will not burn. Blank runs should be made to determine the correction for the combustion of the gelatine capsule.

The heating value of the lighter oils, gasoline, etc., can be obtained by use of the constant pressure, flow-type calorimeter. A special lamp is provided for this purpose with a tank for the fuel. This is mounted on a balance so that the weight of fuel burned can be obtained accurately. This method, however, is not considered standard.

PRODUCTS OF COMBUSTION

The importance of this topic cannot be overestimated since, by the experimental analysis of the gases resulting from the combustion of industrial fuels used for whatever purpose, together with temperature measurements, there may be learned the magnitude of heat losses in the operation of steam boilers, internal combustion engines, gas producers, and all appliances for using fuel either for power or heat. Further, a knowledge of the subject is necessary in order to determine theoretical efficiencies of ideal cycles of internal combustion engines.

Industrial utilization of fuel is accomplished by the rapid chemical reaction between the combustible elements of the fuel and the oxygen contained in air supplied for combustion. The immediate result is heat which elevates the temperature of the combining elements. After combustion occurs in the furnace, or engine cylinder, all of the heat value of the fuel is dissipated in three directions: first, by radiation and conduction during combustion to surrounding objects as boiler tubes, cylinder wall, etc.; second, to dissociated gases which may later combine with a further evolution of heat; and, third, to an increase of energy of the products of combustion over the energy of the elements entering the reaction, this energy being in the form of increase of temperature or increase of enthalpy or both.

With heat-engines or industrial furnaces, the aim is to utilize the heat value of the fuel, especially that large part of it appearing as energy in the products of combustion, and, except where a so-called "reducing flame" is desired, to avoid losses from dissociated gases.

Since it is inevitable that the products of combustion must leave the heat utilization appliance with some elevation of temperature above the atmospheric level, the resulting heat loss will be least when the products of combustion have a minimum weight per unit of fuel; and this weight can be controlled only by supplying air in regulated amount, and properly diffused throughout the fuel, to insure complete chemical reaction. If the air is not properly diffused, its oxygen does not combine, molecule for molecule, entirely with the combustible elements of the fuel; as a result the exhaust gases will contain combustible constituents.

In this text the term **products of combustion** means the mixture of gases and vapors resulting from the complete combustion of a fuel with any amount of air. The term **exhaust gases** means the mixture of gases and vapors actually discharged from a fuel utilizing appliance of whatever kind. It is to be noted that exhaust gas may contain combustible constituents or it may be entirely products of combustion.

Determinations from experimental results must be prefaced upon the following:

1. Constituents of the fuel;
2. Constituents of air;
3. Constituents of the exhaust gas;
4. Temperature of exhaust gas and atmosphere;
5. Reaction equations between 1 and 2 for exact proportioning showing the exact amount of air needed per unit of fuel, and relations between 1, 2, and 3 showing the actual amount of air supplied, products of combustion, unburned combustile, etc., per unit of fuel;
6. The specific heats of air and exhaust gases.

Item 1 has already been considered under pages 142 to 152, and it has been shown that the combustible elements of fuels are: carbon, hydrogen, and sulfur, of which carbon and hydrogen are counted upon to furnish heat of combustion.

Air. The average analysis by volume, as given by Haslam and Russell,* is:

	Per Cent
Nitrogen	78.14
Oxygen	20.92
Argon	0.90
Carbon dioxide	0.04

the water vapor content, of course, varies.

* Fuels and Their Combustion. This analysis omits the rare gases of small volume

It will be observed that the percentages of nitrogen and oxygen are so large as to make the other constituents negligible as far as volume is concerned. Argon is an inert gas and, in combustion processes, passes through with the nitrogen unchanged. The apparatus used for analyzing exhaust gases discloses the argon with the nitrogen, so it will be assumed that the nitrogen content of air is, in per cent, the sum of the nitrogen and argon percentages, that is,

$$78.14 + 0.90 = 79.04.$$

The ratio of the nitrogen volume to the oxygen volume, in air, is a figure very important to combustion calculations, and should be memorized. Using the approximate figures, $N_2 = 79.1$ per cent and $O_2 = 20.9$ per cent.

$$\text{Proportion, by volume, of nitrogen to oxygen in air} = \frac{79.1}{20.9} = 3.78.$$

Consequently, the proportion, by volume, of air to the oxygen contained in it is $(3.78 + 1):1$, or 4.78. It is only necessary to remember the number 3.78 to recall the proportion of air to oxygen. The proportion of air to nitrogen content is, likewise, $4.78:3.78$.

The Constituents of Exhaust Gas. Carbon, completely combined with oxygen, forms carbon dioxide; and hydrogen forms water vapor, or steam. Sulfur forms sulfur dioxide, which, however, is generally so small in amount as to be negligible as far as these determinations are concerned. If sulfur dioxide exists in the exhaust gas, analysis usually discloses it as carbon dioxide. For present purposes, we shall neglect sulfur entirely.

When air is not properly diffused with the fuel, or in the presence of a reducing flame in an industrial furnace, or in the event of a deficiency of air supply in a heat generator, there may be carbon monoxide in the exhaust gas. Carbon monoxide, being combustible with a good heat value (page 154) represents a heat loss. Further, there may be free hydrogen, methane, and, in the combustion of oils, there may be hydrocarbons in small but important amounts. Carbon monoxide is generally taken as an index of incomplete combustion. For the best efficiency, none should appear, but due to poor diffusion, it may be tolerated to 0.1 to 0.2 per cent of the exhaust gas volume. If the carbon monoxide is low, it generally is concluded that no hydrogen or hydrocarbon escapes combustion, but this is not always the fact.

Since an excess of air over that required by the chemical reaction (or the "theoretical air") is necessary to avoid incomplete combustion, the

exhaust gas will contain, besides the products of combustion mentioned, free oxygen. As in air, the main constituent is nitrogen, and the complete list is:

Carbon dioxide
Oxygen
Nitrogen
Water vapor and Carbon monoxide

the first two of which are directly determined, and the third, by difference. The water vapor content of the exhaust gas is usually calculated from the fuel analysis, but it may be found by test of a sample of exhaust gas. A test for carbon monoxide is always made, because its presence even in small quantities indicates appreciable loss to incomplete combustion, and its absence indicates good diffusion of air with fuel. In addition, there may be determined by exhaust gas analysis

Hydrogen
Methane
Hydrocarbons (illuminants)

when combustion is incomplete, but the existence of these gases after combustion in a good fuel-burning equipment is rare.

Before setting up reaction equations for the different fuels and deducing relations between the results of exhaust gas analyses and the air-fuel and products quantities to be determined, certain physical constants and laws will be stated.

Combustion calculations are based upon atomic weights of the elements, the laws governing chemical reactions, Avogadro's law, the laws of Boyle and Charles, Dalton's law, and some elementary thermodynamics. These are briefly recapitulated as follows:

Atomic and Molecular Weights. In the table, page 172, there are given approximate and more exact values of atomic weights of the elements entering combustion of fuels.

For combustion calculations, it is sufficiently accurate to take the approximate atomic weight. The molecular weights then are:

Carbon	C	12
Hydrogen	H_2	2
Nitrogen	N_2	28
Oxygen	O_2	32

Element	Symbol	Atomic Weight	
		Exact	Approximate
Carbon.................	C	12.00	12
Hydrogen...............	H	1.008	1
Sulfur..................	S	32.07	32
Nitrogen...............	N	14.01	14
Oxygen................	O	16.00	16

It is important to note that a molecule of carbon is conventionally represented as having one atom, while molecules of hydrogen, nitrogen, and oxygen each have two atoms.

The molecular weight of a compound is found by adding the weights of the atoms of which its molecule is composed. Thus, the molecular weight of

$$\text{Carbon dioxide, } CO_2, \text{ is } \quad 12 + 32 = 44$$
$$\text{Carbon monoxide, } CO, \text{ is } 12 + 16 = 28$$
$$\text{Steam, } H_2O, \text{ is } \qquad\quad 2 + 16 = 18$$
$$\text{Methane, } CH_4, \text{ is } \qquad 21 + \; 4 = 16$$

and so on.

Avogadro's Law. *In a given volume of any gas, there is always the same number of molecules, regardless of the nature of the gas and of the number of atoms at each molecule, provided the conditions of pressure and temperature are constant.* This law is fundamentally necessary to combustion calculations and it should be thoroughly understood with all its implications.

It follows from Avogadro's law that a given volume of any gas weighs an amount proportional to its molecular weight. Thus, comparing a cubic foot of oxygen with an equal volume of hydrogen, each gas having the same pressure and temperature as the other, the oxygen weighs 16 times as much as the hydrogen, since the ratio of the molecular weights of these gases is $32:2 = 16$. This could only be true because the unit spaces separately occupied by the two gases contain the same number of molecules.

Taking as an example the reaction between oxygen and carbon monoxide

$$2CO + O_2 = 2CO_2$$
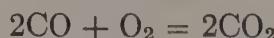
it is to be noted first that this states that two molecules of CO combine with one of O_2 to make two molecules of CO_2. Since the equation is for molecules, it equally well represents volumes, following Avogadro's Law, and may be stated, two unit volumes of CO combine with one of O_2 and make two unit volumes of CO_2. Three unit volumes enter for combustion and the product is two unit volumes; that is, if the pressure-temperature conditions are the same when the volumes are measured, there is a shrinkage equal to two-thirds the volume of the original gases. Shrinkage usually results from the combustion of fuels.

If the volume of CO represented in the reaction is assumed to be that corresponding to 28 lb. of CO (that is, a weight of CO, in pounds, equal numerically to its molecular weight), then the oxygen entering the reaction weighs 16 lb., and the weight of carbon dioxide formed is $28 + 16 = 44$ lb. The reaction may be written, and the weights and volumes indicated, as follows:

$$CO + 0.5 \, O_2 = CO_2$$
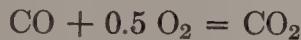
$$28 \text{ lb.} + 16 \text{ lb.} = 44 \text{ lb.}$$

$$1 \text{ volume} + \tfrac{1}{2} \text{ volume} = 1 \text{ volume}$$

$$\text{Volume shrinkage} = 1 \div 1.5 = \text{two-thirds.}$$

At this point the student is recommended to calculate answers to the following:

Exercises. (a) If a cubic foot of hydrogen weighs 0.00562 lb. at 14.7 lb. per sq. in. absolute pressure and 32° F., what is the weight of a cubic foot of oxygen? of nitrogen? of carbon monoxide? of methane? (b) For the following, use the answers to (a). Under the standard conditions, 14.7 lb. and 32° F., how many cubic feet are occupied by 2 lb. of hydrogen? by 32 lb. of oxygen? by 28 lb. of nitrogen? by 28 lb. of carbon monoxide? by 16 lb. of methane? (c) Write the reaction for carbon and oxygen to form carbon dioxide. Indicate all weights and volumes of air and of products as illustrated in the text for carbon monoxide, starting with a weight of carbon equal to its molecular weight. (*Note:* the volume of solid carbon is negligible compared with the gas volumes.) Using the method employed for answer to (b), calculate the volumes of air and products in cubic feet. (d) Same as (c) for hydrogen, assuming the resulting H_2O to have the properties of a gas. (e) Same as (d) for methane.

The mol is a unit of measure very useful in combustion calculations. When understood, it aids simplicity and directness, and eliminates effort of memory or reference for physical constants. By the methods here outlined, any problem in combustion may be solved with a minimum of such effort.

In the exercises just given, the answers to (b) show that the volume occupied under standard pressure-temperature conditions of various gases is 359 cu. ft. regardless of the constitution of the gas molecules, when the amount of each gas, in pounds, is numerically equal to its molecular weight. This quantity of material is the unit known as the mol. (This is the "pound-mol." In physical chemistry, the "gram-mol," similarly defined, is in common use.) A mol of carbon thus weighs 12 lb. and

> 1 mol of H_2 is a quantity weighing 2 lb.
> 1 mol of O_2 is a quantity weighing 32 lb.
> 1 mol of CO_2 is a quantity weighing 44 lb.

A mol of each of these gases at 14.7 psi. and 32° F. occupies 359 cu. ft.

It is necessary to conceive the mol unit as one of weight, varying in magnitude according to the molecular structure of the material to which it is applied as a measure. It helps the student to form a clear conception if he concentrates on the idea of a mol of gas as a unit of volume equal to 359 cu. ft., under standard conditions, regardless of the nature of the gas. It then follows that the weight of this volume, in pounds, varies with the molecular weight of the gas. In this text, the word "mol" always implies a weight if a solid or liquid is referred to; if a gas is referred to, the student should think first of volume, and then of weight. When it is desired to emphasize the significance of the mol as a unit of volume, the word "mol-volume" will be used.

That the volume of a mol of gas is constant may be demonstrated directly from Avogadro's law.

Let d and d' be the densities of two different gases in pounds per cubic foot; V and V' the specific volumes in cubic feet per pound, both d and V being at standard pressure-temperature condition, and let m and m' be the molecular weights of the two gases.

Then, as previously demonstrated,

$$\frac{d}{d'} = \frac{m}{m'}.$$

Since the specific volume is the reciprocal of density, this becomes

$$\frac{V'}{V} = \frac{m}{m'}$$

from which

$$V'm' = Vm.$$

Since the product Vm is the number of cubic feet occupied by a weight of gas in pounds numerically equal to its molecular weight, Vm is the volume of a mol, and equals the volume, $V'm'$ of a mol of any other gas.

A convenient relation following from this characteristic of the mol-volume is

The specific volume of any perfect gas at 14.7 lb. and 32° F., in cubic feet per pound, equals 359 divided by its molecular weight; its density, in pounds per cubic foot, equals its molecular weight divided by 359.

These relations apply strictly only to the so-called "perfect gases" and are not true for vapors. Fortunately the imperfect gases or vapors to be dealt with in combustion follow closely the laws of perfect gases. Even steam, as a product of combustion or as a constituent of air or fuel, can be dealt with as a perfect gas for the reason that its pressure, under these conditions, is very low. (See Dalton's law, following.)

Boyle's and Charles' Laws, combined, are familiarly stated thus:

$$PV = RT$$

in which P = pressure in pound per square foot absolute;
$\quad\quad V$ = volume of 1 lb. of the gas in cubic feet;
$\quad\quad T$ = temperature, degrees absolute, F.;
$\quad\quad R$ = 53.4 for air.

R being a constant for a given gas, this law states that:

(a) If the pressure is constant, the volume of the gas varies directly with its temperature.

(b) If the temperature is constant, the volume of the gas varies inversely with its pressure.

These two laws enable us to correct a gas volume measured under any conditions of pressure and temperature to standard conditions.

Let m denote the molecular weight of a gas. Then multiplying each side of the equation, $PV = RT$, by m,

$$P(Vm) = (mR)T.$$

Vm is the volume of a mol of the gas and its value may be figured using data for any gas. Thus for hydrogen, $R = 772$, and $m = 2$. At 14.7 lb. and 32° F.,

$$Vm = \frac{2 \times 772 \times 492}{144 \times 14.7} = 359 \text{ cu. ft.}$$

and at 14.7 lb. and 60° F.,

$$Vm = 359 \times \tfrac{520}{492} = 380 \text{ cu. ft.}$$

It is to be noted that since (Vm) is a constant in the equation $P(Vm) = (mR)T$ for given values of P and T, regardless of the nature of the gas, then mR also must be constant for all gases. mR is called the **Universal Gas Constant** and its value is 1544.

The careful student will not dismiss the symbols R and mR with the idea that they represent only numbers, but will seek their physical significance. Analysis will show that the quantity, R, is the number of foot-pounds of work done by 1 *lb.* of *a given* gas per degree rise in temperature, when heat is added to it at constant pressure. mR is the number of foot-pounds of work done by 1 *mol* of *any* gas per degree rise in temperature, when heat is added to it at constant pressure. This interpretation is necessary to an intelligent use of specific heat relations, to be outlined later in connection with internal combustion engine efficiencies.

The heat equivalent of the Universal Gas Constant is

$$\frac{mR}{778} = \frac{1544}{778} = 1.985 \text{ B.t.u. per mol per degree F.}$$

Dalton's Law applies to mixtures of gases such as air, fuel gas, products of combustion, and may be used even though the mixture contains some vapors, such as humid air. A statement follows:

In a gas mixture occupying a given volume, the pressure of each constituent gas is the same as though it occupied the volume alone, and the pressure of the mixture equals the sum of the pressures of the constituent gases.

An important corollary to this law, which will not be proved here, is

In a mixture of gases the partial pressure of any constituent gas is to the pressure of the mixture as the per cent by volume of that gas is to 100 per cent. The partial pressures of any two constituents have the same ratio as their volume percentages.

The meaning of "per cent by volume" in this statement, and in refer-

ence to gas analyses in general, should be entirely clear to the student before proceeding. By this expression is meant the number of unit volumes of any constituent gas, per hundred of the mixture, that would be occupied by that gas, if removed from the mixture, and *changed to the same pressure as that of the mixture*, temperature remaining unaltered. It must not be forgotten that, although a constituent gas of a mixture has a pressure *less* than that of the mixture, the volume of the constituent gas is the *same* as that of the mixture. Consequently, the term "per cent by volume" would have no significance unless the volume referred to were reduced to the same physical conditions as those for the percentage basis.

For example, assume air to consist of 79 per cent N_2 and 21 per cent O_2. Assume 100 cu. in. of this air and its pressure to be 14.7 psi. In the air, the

partial pressure of the O_2 = 14.7 × 0.21 = 3.09 lb.

partial pressure of the N_2 = 14.7 × 0.79 = 10.61 lb.

but each of these gases fills the whole space of 100 in. If the oxygen were separated from the nitrogen and the pressure of each of the two gases changed to 14.7 lb., the volume occupied by the O_2 would be 21 cu. in., and by the N_2, 79 cu. in., that is, the O_2 would be 21 per cent, and the N_2 79 per cent of the mixture (air).

Average Molecular Weights and Physical Constants of Mixtures. The average molecular weight of a mixture of gases may be calculated thus:

Multiply the per cent by volume of each gas in the mixture by its molecular weight; add all of these products and divide by 100; the quotient is the average molecular weight.

For example, taking air to consist of 79.1 per cent N_2 and 20.9 per cent O_2, in 100 mol-volumes of air

Weight of N_2, 79.1 × 28 = 2215
Weight of O_2, 20.9 × 32 =   669
____

Sum   2884

Average molecular weight 2884 ÷ 100 = 29.0 (approximately) *

Knowing the average molecular weight of a mixed gas, its constants may be found from the relations given previously. Thus, for air at 14.7 lb. per sq. in. and 32° F.

* The exact value is given authoritatively as 28.97. The reason why the calculation here does not agree is because the greater molecular weights of argon, carbon dioxide, etc., have been neglected. The value, 29.0, is close enough for combustion calculations.

$$\text{Specific volume} = \frac{359}{29.0} = 12.4 \text{ cu. ft. per lb.}$$

$$\text{Density} \quad = \frac{29.0}{359} = 0.808 \text{ lb. per cu. ft.}$$

$$R = \frac{1544}{29.0} = 53.4.$$

We are now ready to calculate the quantity of air required for, and the products from, the combustion of a given fuel. Consider, first, the elements carbon and hydrogen, the principal combustible constituents of all fuels. Remembering that when oxygen from air is used for combustion, 3.78 times as much nitrogen must be supplied with the oxygen, and keeping in mind that the equation for a chemical reaction represents not only molecules, but mol-volumes as far as gases are concerned; the reaction for carbon with just sufficient air for combustion ("Theoretical air") is

$$C + O_2 + 3.78N_2 = CO_2 + 3.78N_2,$$
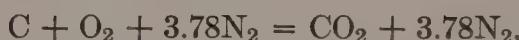
and may be interpreted as 1 mol of carbon combined with 1 mol-volume of oxygen, accompanied by 3.78 mol-volumes of nitrogen, producing 1 mol-volume of carbon dioxide and 3.78 of nitrogen. (*Note:* The mol of carbon, being in solid form, has a negligible volume compared with air or products.) The coefficient before each chemical symbol indicates the number of mols of material represented by that symbol; the coefficients, 1, may, of course, be omitted. There are $1 + 3.78 = 4.78$ mols of air required to burn 1 mol of carbon. The weight of the air is $1 \times 32 + 3.78 \times 28 = 138$. This can also be calculated if the molecular weight of air is taken at 29.0; 1 mol of it then weighs 29.0 lb., and 4.78 mols, $4.78 \times 29.0 = 138$ lb. The weight of the carbon entering the reaction is 12 lb.; consequently, the weight of air required per pound of carbon is $138 \div 12$ or 11.5 lb. This is called the theoretical air-fuel ratio.

The volume of the products of combustion per pound of combustible is readily calculated from the volume of a mol, 359, and equals

$$\frac{359 \times 4.78}{12} = 143 \text{ cu. ft. per lb. of carbon.}$$

Similarly, for hydrogen:

$$H_2 + 0.5O_2 + 0.5 \times 3.78N_2 = H_2O + 1.89N_2.$$
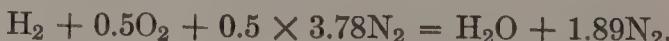
As it takes 2.39 mols of air to burn 1 mol of hydrogen, the theoretical air-fuel ratio is $2.39 \times 29.0 \div 2 = 34.6$ lb. per lb. of hydrogen.

Let us now study a method of calculating the percentages of the products of combustion as they would appear under Test Analysis. So far, theoretical air only has been considered; in actuality, there is generally excess air, and the ratio of the air actually used to the theoretical amount is called the excess coefficient. Let X designate this ratio. Considering the combustion of carbon first, the following reactions are written for, first, theoretical air, second, excess coefficient of 1.5, third, excess coefficient of 2.0.

$$C + O_2 \quad + 3.78N_2 = CO_2 \quad + 3.78N_2$$

$$C + 1.5(O_2 + 3.78N_2) = CO_2 + 0.5O_2 + 5.67N_2$$

$$C + 2.0(O_2 + 3.78N_2) = CO_2 + O_2 + 7.56N_2.$$

In the first reaction, the products consist of 1 mol-volume of CO_2 and 3.78 of N_2, making a total of 4.78. The percentage by volume in the exhaust gas are

$$CO_2 = \frac{1}{4.78} = 20.9\% \qquad N_2 = \frac{3.78}{4.78} = 79.1\%.$$

In the second reaction, the products contain 0.5 mol-volume of free oxygen, and there are $1 + 0.5 + 5.67 = 7.17$ mol-volumes of products. Therefore,

$$CO_2 = \frac{1}{7.17} = 13.9\% \qquad O_2 = \frac{0.5}{7.17} = 7.0\% \qquad N_2 = \frac{5.67}{7.17} = 79.1\%$$

And the last reaction shows

$$CO_2 = \frac{1}{9.56} = 10.45\% \qquad O_2 = \frac{1}{9.56} = 10.45\% \qquad N_2 = \frac{7.56}{9.56} = 79.1\%.$$

It is to be observed that the percentage of nitrogen in the products in each case is 79.1, and that the sum of the percentages of carbon dioxide and free oxygen is 20.9. The addition of an atom of carbon to a molecule of oxygen does not change the original space occupied by the oxygen gas; consequently it makes no difference how much oxygen combines with carbon and how much remains free in so far as the total volume of oxygen in the form of O_2 or CO_2 in the products is concerned. But it is to be noted that, as the excess air is increased, the per cent of CO_2 in the products decreases; consequently the percentage of this gas is a practical index of the amount of air supplied, and is so used in experimental analyses.

Considering hydrogen next, and using again excess coefficients of 1, 1.5, and 2.0

$$H_2 + 0.5O_2 \quad + 1.89N_2 = H_2O \qquad + 1.89N_2$$

$$H_2 + 1.5(0.5O_2 + 1.89N_2) = H_2O + 0.25O_2 + 2.84N_2$$

$$H_2 + 2.0(0.5O_2 + 1.89N_2) = H_2O + 0.5O_2 \quad + 3.78N_2.$$
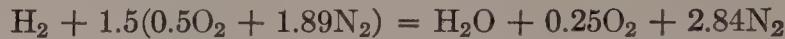
When exhaust gas is to be analyzed, samples of it are usually collected in contact with water; a sample, therefore, becomes 100 per cent humid before the analysis, no matter what its H_2O content as a result of combustion. The apparatus for analysis is adapted to show the constituents of the exhaust gas on a "dry" basis, that is, exclusive of water vapor. Consequently, a test of the products of combustion from a reaction as shown by the first equation would disclose only nitrogen, and, from the second and third reactions, only oxygen and nitrogen. In percentages, the exhaust gases would be

$$O_2 \qquad = 0.0\% \qquad\qquad N_2 \qquad = 100\%$$

$$O_2 = \frac{0.25}{3.09} = 8.1\% \qquad\qquad N_2 = \frac{2.84}{3.09} = 71.9\%$$

$$O_2 = \frac{0.50}{4.28} = 11.7\% \qquad\qquad N_2 = \frac{3.78}{4.28} = 78.3\%$$

It is to be noted, however, that no matter how much air is used, with complete combustion of hydrogen, 1 mol of water vapor results from 1 mol of hydrogen. There are thus formed 18 lb. of water vapor from 2 lb. of hydrogen, or water of combustion weighs nine times the amount of the weight of hydrogen entering combustion.

If the water of combustion is not condensed in the products, then its percentage by volume (using the whole volume, including that of the vapor, as a basis), for the three reactions is, respectively,

$$\text{First,} \quad \text{per cent } H_2O = \frac{1}{1 + 1.89} = 34.6\%$$

$$\text{Second, per cent } H_2O = \frac{1}{1 + 0.25 + 2.84} = 24.4\%$$

$$\text{Third,} \quad \text{per cent } H_2O = \frac{1}{1 + 0.50 + 3.78} = 18.9\%.$$

The partial pressure of H_2O taking the mixture pressure at 14.7 lb. per sq. in. is $0.346 \times 14.7 = 5.1$ lb., $0.244 \times 14.7 = 3.6$ lb., and $.189 \times 14.7 = 2.8$ lb., respectively. By reference to the steam tables it is found that the saturation temperatures, corresponding to these pressures, are respectively 165, 149, and 139° F. If the products in any of the three cases fall to a lower temperature than thus indicated, H_2O vapor will condense. The proportion by weight is unchanged.

The condensation temperatures and "dew points" arrived at in the last paragraph are true assuming moisture-free air to be used for combustion. Humidity in the air will, of course, increase the H_2O content of the products, as will also water contained by solid or liquid fuels and water vapor by gaseous fuels.

Calculations from exhaust gas analysis from any fuel should always be preceded by a reckoning of the *Theoretical Air for Combustion*.

Coals and Oils. From the analysis of the fuel, the carbon and hydrogen contents are known. If the fuel contains no oxygen, the following relation may be used.

$$\text{Air required} = 11.5C_t + 34.6H_t$$

in which C_t = total carbon, in pounds, per pound of fuel;
H_t = total hydrogen, in pounds, per pound of fuel.

The equation gives the weight of air, in pounds, required to burn 1 lb. of fuel.

If the fuel contains oxygen, the equation becomes

$$\text{Air required} = 11.5C_t + 34.6\left(H_t - \frac{O_t}{8}\right),$$

in which O_t is the total oxygen content of the fuel, in pounds, per pound of fuel. The reasoning here is that whatever oxygen is in the fuel may combine with some of the hydrogen; the ratio, by weight, of oxygen to hydrogen in H_2O is 16:2 or 8; consequently, the oxygen in the fuel can combine with one-eighth its weight of hydrogen and that much of this combustible need not receive oxygen from the air.

Fuel Gases. The air-fuel ratio is usually quoted in terms of volumes instead of weights, since fuel gas is metered by volume. A convenient method is illustrated by the use of the following table in which the first two columns give the gas analysis.

In the table, page 182, each percentage in column (2) may be considered as the number of mol-volumes of the corresponding constituent in

DATA FROM TEST OF BALTIMORE CITY GAS

Higher heat value of gas, by calorimeter, 496 B.t.u. per cu. ft. at 14.7 lb. per sq. in. and 60° F., 100% humid.

(1)	(2)	(3)	(4)	(5)	(6)
Constituents of Gas	Per Cent by Volume	Number of Mols, per Hundred Mol-Volumes of Fuel			Weight of Constituent in Pounds
		of C	of H_2	of O_2	
CO..........	15.5	15.5	....	7.75	434.0
H_2..........	43.4	....	43.4	....	86.8
CH_4.........	22.7	22.7	45.4	....	363.2
C_2H_4*.......	3.5	7.0	7.0	....	98.0
O_2..........	0.5	....	....	0.50	16.0
CO_2........	3.3	(Inert, need not be counted)			145.2
N_2..........	11.1				310.8
Total, per hundred mol-volumes of fuel......		45.2 mols of C = c	95.8 mols of H_2 = h	8.25 mols of O_2 = g	1454.0

* The symbol C_2H_4 is used to represent "illuminants" embracing various unsaturated hydrocarbons which are not separately distinguished in the test analysis.

one hundred mol-volumes of fuel. Column (3) is found by multiplying the subscript of C in column (1) by the corresponding figure in column (2). Column (4) is found by multiplying the subscript of H in column (1) by the corresponding figure in column (2) and dividing the product by two; and similarly for column (5). The totals of these columns then give the number of mols of carbon, the number of mol-volumes of hydrogen and of oxygen, respectively, that would result if the combustible constituents of the fuel were dissociated into their elements. These quantities will be referred to as c, h, and g, respectively. Since it takes one mol-volume of oxygen to burn one mol of carbon, one-half a mol-volume of oxygen to burn one of hydrogen, and since 100 mols of fuel already contain g mols of oxygen, the oxygen required from the air equals $c + 0.5h - g$. Consequently, the volume ratio of air to oxygen in air being 4.78,

$$\text{Air required} = 4.78(c + 0.5h - g) \div 100$$

in terms of cubic feet of air per cubic foot of fuel.

$$\text{Air required} = 4.78(45.2 + 0.5 \times 95.8 - 8.25) \div 100 = 4.05 \text{ cu. ft.}$$

Column (6) is obtained by multiplying each item of column (2) by the molecular weight of the corresponding constituent; the product is the number of pounds of each constituent in one hundred mol-volumes of fuel. The sum of column (6) is the weight, in pounds, of one hundred mol-volumes of fuel. Therefore, the average molecular weight of the fuel is the sum of column (6) divided by 100. For the cited data, this is 14.54.

Air-fuel ratio by weight may readily be calculated if the ratio by volume is considered to be in terms of mol-volumes of air per mol-volume of fuel. In the volume ratio may be introduced the weight in pounds of one mol of air and of one mol of fuel, thus

$$\text{Air-fuel ratio, by weight} = \frac{\text{Air-fuel ratio by volume} \times 29.0}{\text{Average molecular weight of fuel}}.$$

The tabulated analysis is for a fuel gas exclusive of the H_2O content. The heat value of the gas is always quoted at 60° F. and 100 per cent humidity, but the gas, as used, may have any other humidity than this. So long as the air used for combustion has the same humidity as the fuel, the theoretical air-fuel ratio as calculated above is correct, because the effect of the humidity then is the same on both air and fuel, that is, it lowers the partial pressure of the oxygen and nitrogen of the air and of the fuel gas constituents as shown, to the same extent in each, without changing the volume or weight ratio of dry air to dry fuel gas for complete combustion.

If, however, the air has different humidity from the fuel, the air-fuel ratio, as just calculated, must be corrected. Suppose, for example, that the fuel is burned at 85° F., with 100 per cent humidity, and that the air used for combustion is at the same temperature and with 40 per cent humidity. From the steam tables, the saturation pressure at 85° is 0.60 lb. per sq. in.; the pressure of the dry gas is then $14.7 - 0.60 = 14.1$ lb. The pressure of the dry air used for combustion is $14.7 - 0.40 \times 0.60 = 14.46$ lb. One mol of air at this pressure would fill a volume equal to $14.1 \div 14.46$ of what it would fill at 14.1 lb., that is, at a pressure equal to that of the dry gas. Consequently, the volume ratio of 40 per cent

humid air for the combustion of 100 per cent humid gas (with an analysis on the dry basis, as tabulated) is

$$4.05 \times \frac{14.1}{14.46} = 3.95 \text{ cu. ft. per cu. ft.}$$

In general,

$$\text{Air-fuel-ratio, as used} = \text{Air-fuel ratio on dry basis} \times \frac{14.7 - P_f}{14.7 - P_a},$$

in which P_a and P_f are the partial pressures in pounds per square inch of the H_2O in the air and in the fuel, respectively.

The effect of humidity upon the heat value of fuel gas as used, especially with high gas temperature, is frequently of importance to accuracy, and should not be ignored.

It is sometimes desirable to express the percentages by volume of the constituents of a fuel gas including the H_2O content. Taking the tabulated fuel gas as an example, consider the proportions of its constituents if the gas is under the conditions quoted for heat value, that is, at 60° F. and 100 per cent humid, the mixture of gas and H_2O vapor being at a pressure of 14.7 lb. From the steam tables, the pressure of the H_2O is 0.26 lb. (at 60°); in consequence, the pressure of the tabulated gas constituents is 14.44 lb. The percentage by volume of the H_2O, based upon the dry gas is, by the law of partial pressures, $0.26 \div 14.44 = 1.8$ per cent. The humid gas then has a volume of 101.8 units as compared with 100 volumes of the dry gas. The volume percentages of the gas constituents   y now be calculated on the "wet gas" basis by dividing each percentage on the "dry gas" basis by 101.8, with the following results:

PER CENT BY VOLUME

Gas Constituent	Dry Basis	Wet Basis
CO.	15.5	15.23
H_2.	43.4	42.61
CH_4.	22.7	22.31
C_2H_4.	3.5	3.44
O_2.	0.5	0.49
CO_2.	3.3	3.24
N_2.	11.1	10.91
H_2O.	1.8	1.77
Total.	101.8	100.00

THEORETICAL REACTIONS OF FUELS

It is sometimes desired, when the excess coefficient is either known or assumed, to write the reaction for a fuel of known analysis. From this may be calculated the theoretical exhaust gas analysis and the quantities of each product of combustion per unit of fuel.

Coals and Oils. Since this type of calculation is particularly useful in the analysis of internal combustion engines, data from an oil fuel will be used to illustrate, as follows:

	Per Cent
Carbon, by weight.............	84
Hydrogen, by weight............	15
Inerts.......................	1

Higher heat value, 20,100 B.t.u. per lb.

Expressing the carbon and hydrogen quantities in mols per pound of oil: carbon, $0.84 \div 12 = 0.07$, and hydrogen, $0.15 \div 2 = 0.075$. Assuming the excess coefficient to be 1.25, a convenient way of writing the reaction follows (this is for 1 lb. of oil):

Mols of combustible..$0.07C + 0.075H_2$

Mols of O_2 required..$0.07O_2 + 0.0375O_2$

Mols of N_2 with the O_2 .... $0.1075 \times 3.78N_2$

Excess air .... $0.25 \times 0.1075(O_2 + 3.78N_2)$

Products...... $0.07CO_2 + 0.075H_2O + 0.0269O_2 + 0.5079N_2$
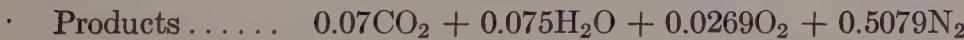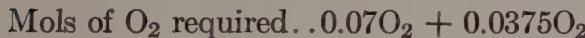
The coefficient of each product represents the number of mols of that product proceeding from the combustion of 1 lb. of oil with 25 per cent excess air. If the corresponding weight in pounds is wished, it is only necessary to multiply each coefficient by the appropriate molecular weight; thus the weight of CO_2 is $44 \times 0.07 = 3.08$ lb.; and the total weight of the products of combustion, in pounds, is

$$44 \times 0.07 + 18 \times 0.075 + 32 \times 0.0269 + 28 \times 0.5079 =$$

$$3.08 + \qquad 1.35 + 0.86 \qquad + 14.22 \qquad = 19.51.$$

The volume in cubic feet of the products of combustion (standard conditions) is the volume of 1 mol, 359, times the number of mols of products, per pound of oil, or

$$359(0.07 + 0.075 + 0.0269 + 0.5079) = 244 \text{ cu. ft.*}$$

It is of convenience, for certain purposes, to express the products of combustion in mols per mol of combustible. For the data under consideration, 1 lb. of oil contains 0.99 lb. of combustible or $0.07 + 0.075 = 0.145$ mol of combustible. Therefore, the weight of oil necessary to supply one mol of its combustible is $1 \div 0.145 = 6.9$ lb. It is now only necessary to multiply the quantities previously found, in terms of mols per pound of oil, by 6.9 to reduce to mols per mol of combustible.

In general, the weight in pounds, of a solid or liquid fuel, necessary to supply 1 mol of combustible in the fuel is

$$1 \div \left(\frac{C_t}{12} + \frac{H_t}{2} \right)$$

in which C_t and H_t are the carbon and hydrogen content of the fuel expressed as fractions.

Gaseous Fuels. Since fuel gas is always measured in units of volume, the products of combustion will be figured in terms of cubic feet, mol-volumes, or pounds per cubic foot or mol-volume of fuel.

Take, as an example, the fuel gas previously cited, the analysis of which, on the dry basis, is:

$$CO = 15.5\% \qquad CH_4 = 22.7\% \qquad O_2 = 0.5\% \qquad N_2 = 11.1\%$$

$$H_2 = 43.4\% \qquad C_2H_4 = 3.5\% \qquad CO_2 = 3.3\%.$$
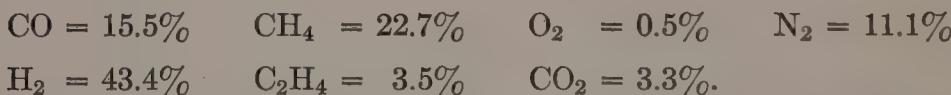
It was shown that if these percentages stand for the number of mols per hundred mols of fuel, the 100 mols of fuel contain 45.2 mols of carbon (exclusive of the carbon in the CO_2), 95.8 mols of hydrogen, and 8.25 mols of oxygen (exclusive of that in the CO_2). Assuming 25 per cent excess air, the reaction may now be written as follows:

Combustible in 100 mols of fuel .... $45.2C + 95.8H_2$

Inerts in 100 mols of fuel..... $3.3CO_2$ $\qquad\qquad\qquad\qquad +11.1N_2$

Theoretical air .... $84.85(O_2 + 3.78N_2)$

$\left\{ \begin{array}{l} \text{Oxygen required from air} = \\ (45.2 + 0.5 \times 95.8 - 8.25)O_2 = 84.85O_2 \end{array} \right\}$

Excess air .... $\underline{0.25 \times 84.85(O_2 + 3.78N_2)}$

Products .... $48.5CO_2 + 95.8H_2O + 21.2O_2 + 412N_2$

* This calculation is on the assumption that the H_2O in the products acts like a perfect gas and does not condense upon reaching its dew point. The assumption does not

In the products there will be as many mols of CO_2 as there were of C in the fuel plus the number of mols of CO_2 in the fuel, and as many mols of H_2O as there were of H_2 in the fuel. The N_2 in the products consists of the N_2 in the air plus that in the fuel.

The coefficients of the constituents of the products in the last line represent the number of mols from 100 mols of fuel. It is only necessary to divide by 100 to reduce to mols per mol, or cubic feet per cubic foot. Thus, in the products, there are 0.485 cu. ft. of CO_2, 0.212 cu. ft. of free O_2, etc., per cubic foot of fuel. The total volume of the products of combustion, per cubic foot of fuel, including H_2O is

$$0.485 + 0.958 + 0.212 + 4.12 = 5.77 \text{ cu. ft.}$$

The theoretical air-fuel ratio is 4.05. With 25 per cent excess air the actual amount of air used is $1.25 \times 4.05 = 5.06$ cu. ft. per cu. ft. of fuel gas. The volume of air plus gas mixed with it, before combustion, is thus $5.06 + 1 = 6.06$ cu. ft., greater than the volume of the products, just calculated to be 5.77 cu. ft.

Shrinkage Factor. The products of combustion from gas fuels are generally less in volume (referred to standard conditions) than the volume of the air and fuel entering combustion, as shown by the preceding calculation. The ratio of these volumes is called the "shrinkage factor" or "coefficient of contraction." For the example given, if none of the H_2O of combustion is condensed, its value is $5.77 \div 6.06 = 0.952$.

Let V_p be the volume in mols of the products from 100 mols of fuel gas, as found by adding the coefficients on the products side of the reaction (as on page 186).

X, excess coefficient.

K, theoretical air-fuel ratio, by volume.

Then,

$$\text{Shrinkage factor is } \frac{V_p}{100(XK + 1)},$$

provided no water of combustion is condensed.

To allow for condensation of H_2O, suppose, for example, that the reaction written on page 186 is for dry fuel supplied with dry air ($X = 1.25$) and that both are at 60° F. before combustion, and that the products, after

lead to error when the volume thus found does not represent an actual physical condition, but is used only for a recalculation at another temperature, above the dew point, or for the purpose of comparison.

combustion, are cooled to 60° F., at a pressure of 14.7 lb. Some of the H_2O of the products condenses in this cooling. At 60°, the partial pressure of the H_2O (saturated) is 0.26 lb. The partial pressure of all of the remaining products is $14.7 - 0.26 = 14.44$ lb. The per cent by volume of the H_2O remaining as steam, based on that of other products, is $0.26 \div 14.44 = 1.8$ per cent. The number of mols of the other products is, from the reaction,

$$48.5 + 21.2 + 412. = 481.7.$$

The number of mols of H_2O, as steam, at 60°, is therefore $0.018 \times 481.7 = 8.7$, and the total number of mols of products including H_2O is $481.7 + 8.7 = 490.4$ per hundred of fuel, or 4.90 volumes per unit volume of fuel. Therefore the shrinkage factor is:

$$\frac{4.90}{1.25 \times 4.05 + 1} = 0.808.$$

CALCULATIONS FROM EXHAUST GAS ANALYSES

The quantities, equations for which are to be presented, are based upon a unit of 1 lb. weight for coals and oils, and upon 1 standard cu. ft. at 14.7 lb. per sq. in. and 32° F. for gaseous fuels. Two different sets of equations· are needed. The following notation will be used for the two cases in common:

D, M, O, N, H = percentages of carbon dioxide, monoxide, oxygen, nitrogen, and hydrogen, by volume, in the exhaust gas, respectively.

V_d = volume of dry exhaust gas in cubic feet per pound of coal or per cubic foot of fuel gas.

V_p = volume of exhaust gas, including H_2O, in cubic feet per pound of coal or per cubic foot of fuel gas.

W_a = weight of air supplied in pounds per pound of coal, or per cubic foot of fuel gas.

W_d = weight of dry exhaust gas in pounds per pound of coal, or per cubic foot of fuel gas.

W_v = weight of H_2O in the exhaust gas, per pound of coal, or per cubic foot of fuel gas.

X = excess coefficient; that is, the ratio of the amount of air supplied to that required for complete theoretical combustion.

Formulas for Coal Combustion. In 100 mol-volumes of the exhaust gas there are $D + M$ mols of carbon since each per cent of CO_2 and CO contains 1 mol of carbon. Therefore there are $100 \div (D + M)$ mols of exhaust gas per mol of carbon in it, exclusive of H_2O, since the values D and M are found experimentally on a basis of dry gas. The volume in cubic feet of the dry exhaust gas, per pound of carbon contained, is this quantity multiplied by the volume in cubic feet of 1 mol of a gas and divided by the weight of a mol of carbon in pounds, or

$$\frac{100}{D + M} \times \frac{359}{12} = \frac{2980}{D + M}.$$

If the fuel contained only carbon, and if all of it appeared as CO_2 or CO in the exhaust gas, this would be the expression for V_d. In the combustion of coal (aside from pulverized coal), some carbon is lost through the grate and in cleaning fires, and some is unaccounted for in the form of smoke or soot. To relate the carbon appearing in the exhaust gas to that of the coal, it is necessary to determine that part of the carbon from 1 lb. of coal which is burned, that is, converted by combustion to either or both of the gases CO_2 and CO. If C_g represents the carbon gasified expressed as a fraction, then

$$V_d = 2980 \frac{C_g}{D + M}.$$

The volume of the products of combustion, including H_2O, per pound of fuel may now be found from V_d. If H_t represents the part of a pound of hydrogen contained in 1 lb. of fuel, there are $H_t \div 2$ mols of hydrogen per pound of fuel and there will be the same number of mols of H_2O of combustion in the products. The volume in standard cubic feet of this H_2O per pound of coal is $359 \times (H_t \div 2)$. If 1 lb. of the fuel contains m pounds of H_2O as moisture, the volume of this H_2O in the products is $359(m \div 18)$. Neglecting the humidity in the air used for combustion, V_p equals V_d plus these two H_2O volumes, or

$$V_p = V_d + 359 H_t \div 2 + 359 m \div 18$$

$$= V_d + 20(9 H_t + m).$$

The quantity, V_p, is useful in the investigation of stack gas velocities. However, for this purpose it must be corrected for existing temperature and pressure.

Before proceeding, the quantity C_g, the weight of carbon gasified per pound of coal, will be considered.

Let C_t represent the total carbon, in pounds, in 1 lb. of dry coal; and
C_a be the weight of the carbon per pound of dry coal lost through the
grates or removed in cleaning fires.

Then, neglecting carbon in smoke or soot,

$$C_g = C_t - C_a.$$

C_t may be estimated from the proximate analysis, and C_a measured by
analysis and total weighed quantity of "ash and refuse" and total weighed
coal quantity from test.

To find the weight of air supplied per pound of fuel, it is assumed that
all the nitrogen in the exhaust gas comes from the air, a very nearly correct
assumption, since coal and oil contain but little nitrogen.   There are
$N \div (D + M)$ mols of nitrogen per mol of carbon in the exhaust gas.   The
air volume corresponding to this nitrogen is $4.78 \div 3.78$ times the volume
of the nitrogen.   The number of mols of air supplied per mol of carbon
in the exhaust gas is

$$\frac{4.78}{3.78} \times \frac{N}{D + M}.$$

Since the weight of 1 mol of air is 29 lb., and of 1 mol of carbon 12 lb.,

$$W_a = \frac{29}{12} \times \frac{4.78}{3.78} \times \frac{N}{D + M} \times C_g = 3.04 \frac{NC_g}{D + M}.$$

The Boiler Test Code of the A.S.M.E. gives another relation for W_a—
namely,

$$W_a = W_d + 8H_t - C_g,$$

which, interpreted, states that the weight of air actually used equals the
weight of dry products plus the oxygen supplied for water of combustion
minus the carbon from the coal.

The weight of the dry products in pounds per pound of dry coal follows
from molal relationships.

$$W_d = \frac{44D + 32O + 28N + 28M}{12(D + M)} \times C_g$$

$$= [11D + 8O + 7(N + M)]C_g \div 3(D + M).$$

Water vapor in the exhaust gas comes from three sources—namely, from
combustion of hydrogen, from moisture in the coal, and from the moisture
in the air supplied.   The first two only need be considered.

If H_t is the weight of hydrogen in 1 lb. of dry coal, the resulting water vapor, if all the hydrogen is burned, will be $9H_t$, since the ratio of weights of H_2O to the H_2 in it is $(2 + 16) \div 2 = 9$. If m is the weight of moisture in the coal per pound of dry coal, then the total weight of H_2O in the exhaust gas, per pound of dry coal is

$$W_v = 9H_t + m.$$

The assumption that all the H_2 is burned is sufficiently accurate under usual operating conditions.

If H_t and m are found from the proximate analysis (see Test 39(c) and (i)) as percentages of the coal as analyzed, they should be divided by 100 minus the percentage of moisture, to reduce to pounds per pound of dry coal.

The weight of carbon incompletely burned, per pound of dry coal, may be calculated from the CO appearing in the exhaust under the assumption that there is no incomplete combustion through unburned hydrocarbons. The method is the same as for the determination of the weight of air. Let W_i be the desired weight. Then

$$W_i = \frac{12MC_g}{12(D + M)}.$$

$$= \frac{MC_g}{D + M}$$

If there is any other combustible than CO in the exhaust gas, as H_2, CH_4, C_2H_4, a corresponding equation may be written

$$W'_i = \frac{nI}{12(D + M)} \times C_g,$$

in which W'_i is the weight of the combustible per pound of coal, I is the per cent by volume of the combustible in the exhaust gas, as determined by analysis, and n is its molecular weight.

To find the excess coefficient, it is necessary to divide the value of W_a, as deduced above, by the expression for the air required as given on page 181; then, neglecting O_2 in the coal,

$$X = W_a \div (11.5C_t + 34.6H_t).$$

It is sometimes convenient to estimate the excess coefficient from the exhaust gas analysis only, when the fuel analysis is unknown. This may be done by the following reasoning.

The total air supplied is represented by the nitrogen in the exhaust gas. The excess air is represented by the free oxygen, except that, if there is carbon monoxide, some of the free oxygen should have been applied to forming carbon dioxide instead, in amount equal to one-half the volume of the carbon monoxide. The difference between the air supplied and the excess air is the air required. Accordingly,

$$\text{Air supplied} = 4.78N \div 3.78$$

$$\text{Excess air} \quad = 4.78(O - 0.5M)$$

$$\text{Air required} = 4.78N \div 3.78 - 4.78(O - 0.5M).$$

The air-supplied quantity divided by the air required is the excess coefficient

$$X = \frac{4.78N}{3.78} \times \frac{1}{4.78N/3.78 - 4.78(O - 0.5M)}$$

$$= \frac{N}{N - 3.78(O - 0.5M)}.$$

It is to be noted that this expression accounts only for the air required to burn the carbon, that is, the carbon in the exhaust gas appearing as CO_2 or CO. The expression for X previously derived counts as "air required" that which is necessary to burn all of the carbon in the coal, whether or not there is a loss of carbon through grate and in cleaning fires.

The formulas above also apply to the **combustion** of **oils** if C_t is substituted for C_g, and if m is left out of the expression for W_v.

Formulas for Gas Combustion. These are deduced upon the assumption that all the carbon in the fuel appears in the exhaust gas analysis, and that none other does. Contrary to this, in the case of internal combustion engines, is the fact that some carbon is left as a deposit in the cylinder, but it is so small compared with the total carbon used as to be negligible. Also, if the lubricating oil burns, it will appear as CO_2 in the exhaust, so care should be taken to avoid this condition. Hydrocarbons in the exhaust, too, will prevent the correct application of the formulas. If the CO in the exhaust is less than 1 per cent, it is very unlikely that there is any hydrogen or hydrocarbon, because CO is the least readily burned of the fuel gas constituents, because of its relatively high ignition temperature.

The following relation will be used in most of the formulas.

Weight or volume of the substance sought, per cubic foot of fuel gas = weight or volume of that substance contained in 1 cubic foot of exhaust gas × volume of exhaust gas per cubic foot of fuel gas.

If an expression for the last-named quantity, which is V_d, be deduced, it will remain only to find the required quantities per cubic foot of exhaust gas.

The following additional notation will be used.

c, h, g = the number of mols in 100 mol-volumes of the fuel gas, of carbon, hydrogen, and oxygen, respectively; counted as in columns 3, 4 and 5 of the table on page 182.

K = Ratio, by volume, of air supplied to gas, that is, the number of cubic feet of air per cubic foot of fuel.

To find V_d. Since there are $D + M$ mols of carbon in 100 of the exhaust gas, and c mols in 100 of the fuel gas,

$$\text{Mols of exhaust gas per mol of carbon in it} = \frac{100}{D + M}.$$

$$\text{Mols of fuel gas per mol of carbon in it} = \frac{100}{c}.$$

By assumption, the carbon is the same in both gases. Hence, by division,

Mol-volumes of exhaust gas per mol-volume of fuel gas

$$= \frac{100}{D + M} \div \frac{100}{c} = \frac{c}{D + M}$$

and, since this is a volume ratio, it also represents the number of cubic feet of exhaust gas per cubic foot of fuel gas, that is, V_d.

$$V_d = \frac{c}{D + M}.$$

The volumes of exhaust gas above are for dry gas.

The volume of air supplied per cubic foot of fuel gas is determined as follows: The oxygen appearing in 100 mol-volumes of the exhaust gas is $D + O + 0.5M$ mol-volumes. The corresponding amount of air is 4.78 times this amount; and the cubic feet of air which supplied this oxygen, per cubic foot of exhaust gas, is $4.78(D + O + 0.5M) \div 100$. Multiplying by V_d gives the cubic feet of air per cubic foot of fuel that came in with the oxygen that *appears* in the exhaust gas. But some oxygen has

disappeared in the form of water. As there are h mols of hydrogen in 100 of the fuel, and as each one combines with half its volume of O_2, $0.5h \div 100$ is the cubic feet of oxygen disappearing as water, per cubic foot of fuel. But some of the oxygen comes from the fuel itself, and we want only that which comes from the air. Subtracting the mols of oxygen in the fuel gives $(0.5h - g) \div 100$. Multiplying by 4.78, we get the corresponding air. Then the cubic feet of air supplied per cubic foot of fuel gas is

$$K = \frac{4.78}{100}[(D + O + 0.5M)V_d + (0.5h - g)].$$

If there is H per cent of hydrogen in the exhaust, too much oxygen has been counted. To allow for this, if the free hydrogen is known, subtract $0.5H$ from the quantity in the first parentheses.

To find W_d. One hundred mols of the dry exhaust gas weigh $44D + 32O + 28M + 28N$ lb. The volume of 100 mols is 100×359 cu. ft. The weight per cubic foot of the dry exhaust gas is therefore

$$\frac{44D + 32O + 28(M + N)}{359 \times 100}.$$

Multiplying this by V_d, and simplifying, we have, very nearly,

$$W_d = \frac{V_d}{9000}[11D + 8O + 7(M + N)].$$

To find W_v. Besides the water of combustion, there is some water vapor in the exhaust due to moisture in the air and fuel gas. These will be disregarded, as they have but a slight effect upon the heat analysis, owing to the fact that latent heat is not lost to the H_2O of humidity.

Since there are $h/100$ cu. ft. of hydrogen per cubic foot of fuel, and since the weight of the resulting H_2O is 9 times the weight of the H_2, we have,

$$W_v = 9 \times \frac{h}{100} \times \frac{2}{359}$$

$$= 0.0005h.$$

If hydrogen in the exhaust has been analyzed,

$$W_v = 0.0005(h - HV_d)$$

since free hydrogen would mean a corresponding decrease of H_2O.

The cubic feet of unburned CO per cubic foot of fuel equals

$$V_i = MV_d \div 100.$$

If hydrogen has been found, it may be expressed similarly.

To find the excess coefficient, the value of K, as deduced above, should be divided by the volume of air required per cubic foot of fuel, as determined on page 183, being equal to $(c + 0.5h - g) \times 4.78 \div 100$. Consequently,

$$X = \frac{100K}{4.78(c + 0.5h - g)}.$$

SPECIFIC HEATS, ENERGY AND ENTHALPY OF GASES

For low temperature ranges the specific heats of air, products of combustion, etc., may be taken as constants, with sufficient accuracy, but as these quantities are functions of temperature, account must be taken of their variation at and beyond about 1000° F.

In this work, use only of molal specific heat will be made, as, in these terms, calculations are greatly simplified. Molal specific heat is the number of B.t.u. necessary to raise 1 mol of a material 1° F., in temperature.

Let k_v = molal specific heat of a gas when heat is added at constant volume;

k_p = molal specific heat of a gas when heat is added at constant pressure;

T = absolute temperature, degrees F.

1.985 = universal gas constant in B.t.u. per mol per degree.

Then

$$k_p = k_v + 1.985,$$

because k_v also represents the internal energy change per mol per degree, and 1.985 is the heat equivalent of the work done, per mol per degree, when heat is added at constant pressure which heat addition equals k_p, or enthalpy.

The specific heat, k_v, may be represented by the general equation

$$k_v = M + BT + CT^2$$

and it must be remembered that this is the *instantaneous* specific heat. In this equation, M, B, and C are constants experimentally determined for the gas. Each constant has the same value for O_2, N_2, and CO, and

different values for other gases. For CO_2, one set of values is assigned when T is less than 2900°, and another set when T is greater.

The specific heat at constant pressure is expressed by the parallel equation

$$k_p = M' + BT + CT^2$$

in which

$$M' = M + 1.985,$$

since

$$k_p = k_v + 1.985.$$

The following table gives values of M, B, and C for various gases, as determined by Goodenough and Felbeck,* which values have been widely quoted and used in engineering practice.

SPECIFIC HEAT CONSTANTS FOR GASES

Gas	M	$1,000 \times B$	$1,000,000 \times C$
O_2, N_2, CO..............	4.945	0.0	0.1200
H_2........................	4.015	0.6667	0.0
CH_4......................	1.474	10.56	0.0
C_2H_4.....................	4.685	6.80	0.0
C_8H_{18} (gasoline)...........	36.342	38.00	0.0
H_2O......................	6.345	−0.276	0.423
$CO_2(T < 2,900)$...........	5.165	3.90	−0.60
$CO_2(T > 2,900)$...........	10.211	0.420	0.0

Values of $1000 \times B$ and $1,000,000 \times C$ are quoted, instead of B and C, to avoid long decimal expressions; the values read from the table must be divided by 1000 and 1,000,000, respectively, before insertion in the specific heat equations.

Specific Heats of Gas Mixtures. The general form of the equation is the same as already given, but the values of M, B, and C must be calculated for each mixture. Let N_1, N_2, etc., be the number of mols of each constituent gas in a mixture containing a total of N mols, that is,

$$N = N_1 + N_2 + N_3, \text{ etc.}$$

and let the subscripts 1, 2, 3, etc., be applied to the symbols M, B, C to represent corresponding gas constants, the constants for the gas mixture having no subscript. Then

* University of Illinois, Engineering Experiment Station Bulletin No. 139.

$$M = (N_1 M_1 + N_2 M_2 + N_3 M_3 + \cdots) \div N$$

$$B = (N_1 B_1 + N_2 B_2 + N_3 B_3 + \cdots) \div N$$

$$C = (N_1 C_1 + N_2 C_2 + N_3 C_3 + \cdots) \div N.$$

The values N_1, N_2, N_3 may be taken from the volume composition in per cents of an air-fuel mixture, or of the products of combustion by analysis, or from the number of mols appearing in a theoretical equation for a chemical reaction.

The internal energy of gas mixtures in B.t.u. per mol equals $\int_0^T k_v dT$.

This is an arbitrary expression by which the total internal energy is counted above absolute zero on the assumption that the specific heat equation holds true for the whole temperature range from zero to T degrees. This, of course, is untrue, but no error results, as the expression is applied always to find the differences between energies at two different temperatures, T_1 and T_2. It is simpler to integrate between zero and each value of T separately than to integrate between T_2 and T_3.

Substituting the value k_v in the integral,

$$\text{Internal energy} = \int_0^T (M + BT + CT^2) dT = MT + BT^2 \div 2 + CT_3 \div 3$$

in B.t.u. per mol of the gas mixture.

The enthalpy of gas mixtures in B.t.u. per mol is derived in the same way, using k_p instead of k_v. Since $M' = M + 1.985$,

$$\text{Enthalpy} = \int_0^T k_p dT = \int_0^T (M + 1.985 + BT + CT^2) dT$$

$$= 1.985T + MT + BT^2 \div 2 + CT^3 \div 3$$

$$= 1.985T + \text{internal energy}.$$

To make these relations clear, a numerical example will be worked. A fuel gas composed, by volume, of

$$40\% \text{ CO, } 10\% \text{ H}_2, \text{ } 50\% \text{ N}_2$$

is burned with 20 per cent excess air. Assuming complete combustion, find the specific heat constants, M, B, and C, and the internal energy and enthalpy at various temperatures of the air-fuel mixture and of the products.

The theoretical reaction may be written as follows:

Fuel .... $40CO + 10H_2 + 50N_2$
Air required...... $25O_2 + 25 \times 3.78N_2$
Excess air .... $5O_2 + 5 \times 3.78N_2$
Products .... $40CO_2 + 10H_2O + 5O_2 + 163.4N_2$
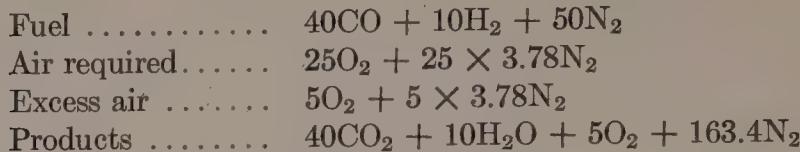
from which there are in the air-fuel mixture:

of CO, O_2, N_2.... $40 + 30 + 30 \times 3.78 + 50 = 233.4$ mols
and of H_2 .... $\underline{10}$ mols

Total .... 243.4 mols

and, in the products mixture, there are

of the O_2 and N_2, $5 + 163.4 = 168.4$ mols
and of H_2O .... 10.0 mols
and of CO_2............... $\underline{40.0}$ mols

Total .... 218.4 mols

From the table of specific heat constants, for the air-fuel mixture

$$M = (233.4 \times 4.945 + 10 \times 4.015) \div 243.4 = 4.907$$

$$B = (233.4 \times 0.0 + 10 \times 0.6667) \div 243,400 = 0.0274 \div 1000$$

$$C = (233.4 \times 0.12 + 10 \times 0.00) \div 243,400,000 = 0.1151 \div 1,000,000.$$

In the application of these constants to the energy equations, it is of assistance to plot a curve of energy in B.t.u. per mol against absolute temperature. Points for this curve will be located at 600°, 1000°, 1500°, 2000°, and 2500°.

The internal energy of the air-fuel mixture at 600° equals

$$4.907 \times 600 + \frac{0.0274}{1000} \times \frac{360,000}{2} + \frac{0.1151}{10^6} \times \frac{216 \times 10^6}{3} = 2958 \text{ B.t.u.}$$

and at 1000°,

$$4907 + 13.7 + 38.3 = 4959 \text{ B.t.u.}$$

A convenient method of making these energy calculations is by ratio, using a base value. In what follows, we shall refer to that term in the energy equation involving T as the first term, the term involving T^2 as the second term, and that involving T^3 as the third term. The "base values" of these terms are at 1000° F. absolute, and are shown in the first column of the following table. Then the first term at any other tempera-

ture, T_2, equals the base value of the first term times the ratio of T_2 to 1000; the second term at T_2 equals its value at 1000° times the square of the ratio T_2 to 1000; and the third term, at T_2, equals its value at 1000° times the cube of the temperature ratio.

	Base Value, 1000°	At 1500°, Base Times	At 2000°, Base Times	At 2500°, Base Times
First term...........	4,907	1.5 = 6,360	2 = 9,814	2.5 = 11,267
Second term...........	13.7	2.25 = 31	4 = 55	6.5 = 89
Third term...........	38.3	3.37 = 129	8 = 306	16.25 = 623
Sum...............	4,959.0	6,520	10,175	11,979

The energy in B.t.u. per mol above absolute zero, at each tabulated temperature, is the sum of the first, second, and third terms under that temperature, as shown.
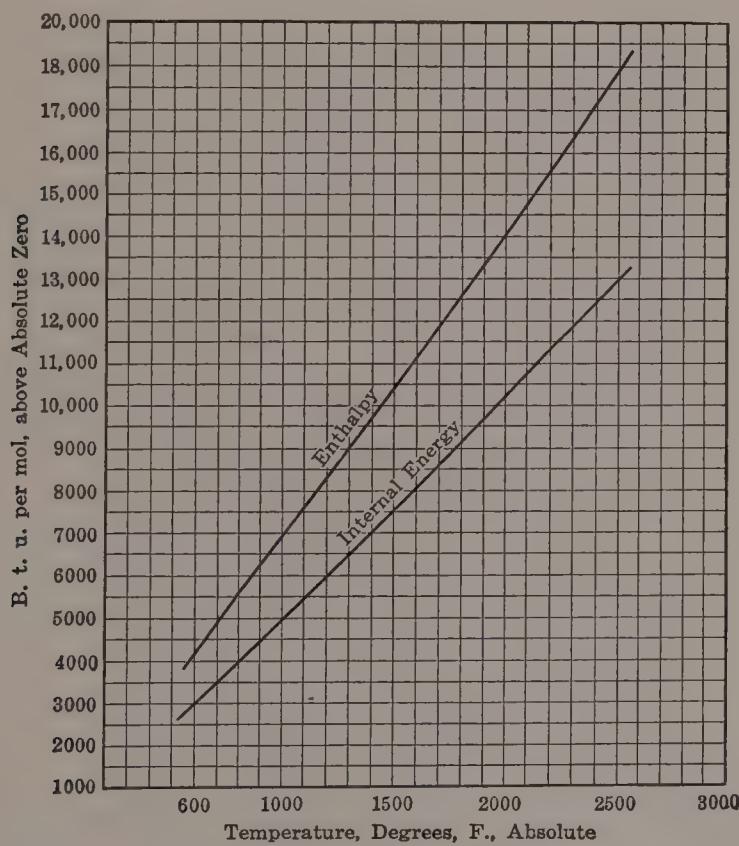
FIG. 91.—Internal Energy and Enthalpy of Air-fuel Mixture.

The enthalpy is now readily obtained by adding the $1.985 \times T$, at each temperature to the internal energy.

$$\text{At } 1000°, \text{ enthalpy } = 4,959 + 1985 = 7,944 \text{ B.t.u.}$$
$$1500°, \text{ enthalpy } = 6,520 + 2977 = 9,497$$
$$2000°, \text{ enthalpy } = 10,175 + 3970 = 14,045$$
$$2500°, \text{ enthalpy } = 11,979 + 4962 = 16,941.$$

These values of energy and enthalpy are plotted against absolute temperature as shown in Fig. 91.

46. The Analysis of Exhaust Gas

Principles. The gas analysis apparatus depends upon the separate absorption of the products of combustion by certain reagents. A measured volume of the gas is brought into contact with one of these reagents which removes the CO_2, but does not take up any of the other constituents. The CO_2 is then determined by measuring the diminution in volume. In this way the volumes of CO and O_2 may also be found. The residue is assumed to be N_2. It is convenient to make the original volume 100 units, as cubic centimeters, in which case the differences are per cents.

Fig. 92.—Gas Sampling Bottle.

The Orsat apparatus in its various forms is in general use for engineering analyses of exhaust gas. It consists of a measuring flask, B', called the burette (see Fig. 93), a distributing tube, T, and three so-called pipettes, DD', OO', and MM', containing the reagents. The gas sample is displaced from the burette into each pipette in turn by filling the burette with water from the bottle B. The right leg of the pipette is completely filled with reagent just before the gas enters; the gas displaces it into the left leg. Rubber bags, R, seal the left legs of the pipettes so that the reagents will not deteriorate through contact with the atmosphere, or other unburned hydrocarbons.

If desired, the apparatus may be provided with a fourth pipette for determining hydrogen.

There are many other types of exhaust gas analysis apparatus, but the most important for simple determinations is the Orsat of which there

are several modifications, especially in the pipettes, for the purpose of making them easier to clean, or for better contact between gas and reagent, and so on. Recording CO_2 analyzers have had a vogue in power plant work. For accurate experimental work, however, the Orsat or modification, manually operated, should be used for convenience and reliability.
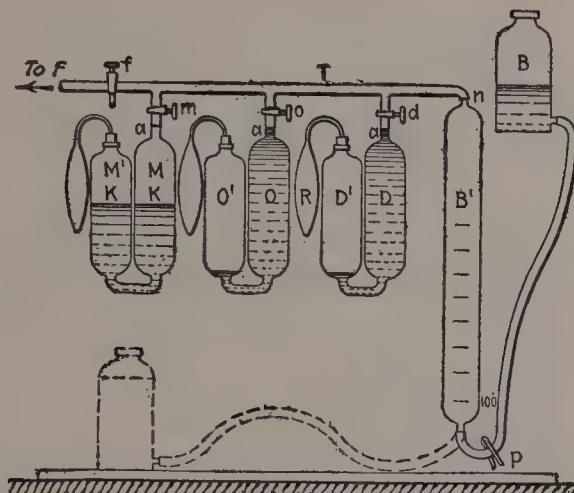
FIG. 93.—Orsat Apparatus.

Reagents for Orsat Apparatus. For CO_2, a 1 to 2, by weight, solution of potassium hydroxide (KOH) or sodium hydroxide (NaOH) in water. The reagent will absorb 40 times its own volume of CO_2.

For O_2, the reagent is a mixture of water, potassium hydroxide, and pyrogallic acid.

This is prepared by dissolving one part, by weight, of KOH or NaOH in two of water, making solution A. Another solution B is prepared by dissolving 1 part of pyrogallic acid, by weight, to three of water. Equal volumes of solutions A and B should be mixed to make the reagent for O_2. The absorption capacity is twice its own volume.

For CO, the reagent is a 3 to 20 solution of cuprous chloride (CuCl) in water. To this should be added just enough strong ammonia to cause a blue color. The absorptive capacity is four times its own volume.

Solutions A and B may be mixed in quantity, A being useful either for CO_2 as mixed or for preparing the O_2 reagent. The CuCl may also be kept in solution without the addition of the ammonia.

Sampling. For use in the apparatus, a small amount of the total gas is led from the main body through a sampling tube. The proper construc-

tion of this tube depends upon whether the gas is flowing in a large flue, as in boiler work, or in a comparatively small pipe, as in gas engine work. For the latter it is sufficient to tap into the exhaust pipe a pipe of about ¼-in. diameter. In a boiler flue, however, the composition of the gas may vary through any section of the current due to stratification, so means should be provided for drawing gas from various parts of the section. One method of doing this is to run the sampling pipe diagonally across the flue, the end of the pipe being sealed and holes being drilled across the length in the flue. The holes should increase uniformly in size from the point of entrance of the sampling pipe to its end, as otherwise more gas will be drawn from the holes in the near end of the pipe. The pipe should enter as nearly as possible to the furnace, but beyond the combustion chamber or last pass in which combustion is taking place. This should be done because the farther the gas is from the furnace the greater is its opportunity to become diluted with air entering through leaks in the brick work or setting.

The sample may be collected with an apparatus such as that shown by Fig. 92, consisting of two 3- and 5-gal. flasks connected as shown. The flask S is used to receive the sample. To start with, it is filled with water. During collection, the water is syphoned into flask R through T' and the gas flows through H into the space evacuated, branch F being closed.

All the air in the tube H should be displaced before taking the sample. This may be done either by raising the flask R above the level of the sampling tube so that water will syphon into the tube H, or by drawing a small charge of gas into flask S in the regular manner and then discharging it through a three-way cock at the end of the branch F. The latter procedure leaves flue gas in the tube H very slightly diluted with air.

The sample should be collected at a uniform rate, and, if more than one sample of the same gas or for the same test is to be analyzed, time periods during which collections are made should be equal in order to get a correct average result. The rate may be kept practically uniform by placing flask R several feet below S and by choking the end of the syphon at t with a sliver of wood. The change in the rate due to the decreasing distance between the water levels in the flasks may then be inconsiderable.

For boiler and gas engine tests, it is well to collect the sample during half-hour or hour periods, and during the collection of each sample to analyze the preceding one. This has the advantage of showing the uniformity of combustion as the test proceeds. It is also well to have a

duplicate sampling outfit, collecting gas at a very slow rate so that the sample will cover the whole test. This may be analyzed at the convenience of the experimenter; the results should check the average of the separate analyses.

(a) **Determination of CO_2, O_2, CO and N_2.** The Orsat apparatus is first connected with a rubber tube to the branch F leading from the sample bottle (see Figs. 92 and 93). The reagents in the pipettes are brought to the levels a, by manipulation of the bottle B, the three-way cock f being closed. This being done, and the cocks, d, o, and m, closed, the water level in the burette is raised to the point n, thus expelling the air previously contained through the three-way cock f which is open to the atmosphere for that purpose. The pinch-cock C on the sampling bottle flue connection is now closed, and the flask R raised so as to put the sample under pressure. The cocks between the sample bottle and the burette are then opened, and the bottle B lowered so that part of the sample will flow into the burette. The gas now in the burette is diluted with air or other gas which was in the distributing tube and in the rubber tube leading to the sampling bottle. For a purer charge, this first one is discarded through the three-way cock, f, and another one brought into the burette, in the same way as the first, which may be accepted for analysis. This charge should be somewhat greater than 100 cc.

Having secured this sample by closing the three-way cock, the bottle B is raised a trifle until the water level in the burette reaches the lowest graduation which marks 100 cc. of gas. As the sample has been compressed by this rise in water level, the pressure should be made equal to atmospheric by opening the three-way cock for an instant, the pinch-cock p being closed so as to maintain the water level at the lowest graduation.

The sample should now be worked back and forth in the KOH solution to remove the CO_2. After this has been done for a few minutes, it should be returned to the burette, care being taken that the reagent is at its original level, a; the cock d should be closed; and a volume measurement of the gas made. This must be done with the sample under the same condition of pressure as when originally measured, that is, atmospheric. For this reason, the water levels in the burette and in the bottle B should be coincident when the reading is made. Before making the reading it is well to allow the walls of the burette to drain for a minute. After the reading the gas is returned to the KOH solution and worked a few more times, then measured again. If the volume is the same, the

result is satisfactory; if not, the process should be repeated until all the CO_2 has been removed as indicated by constant volume. The diminution of volume is the per cent CO_2.

The O_2 and CO are removed and measured in the same way and in the order named. The difference between the sum of the per cents of CO_2, O_2, and CO, and 100 may be taken as the percentage of N_2.

Precautions in Operating. Before using the apparatus, all connecting tubes and cocks should be tested for leaks by putting air in the system under pressure or partial vacuum with the bottle B, and noting whether the volume remains constant as shown by the level in the burette.

Analyses should be made at a uniform temperature of not less than 60° F. During operation, the apparatus should not be exposed to a changing temperature, or placed where drafts or sun may strike it. The corresponding changes in the volume of the sample would give false results.

Water absorbs all of the gases to some extent. To avoid the resulting errors, it is well to use water that has been previously saturated with exhaust gas, so that it will not take up any more. To do this the exhaust gas is caused to bubble through the water used in the sampling flask and in the burette.

If a little coloring matter, such as methyl orange, is added to the water used in the burette, it will aid in the reading of the graduations.

It is well to keep a record of the volumes absorbed by each reagent so that it will be known when their absorptive capacity has been reached. The volume of reagent in a pipette is about 150 cc.

(b) Calculations from Exhaust Gas Analyses. The various quantities obtainable are expressed by formulas derived on pages 189 to 194.

For practice, the student may analyze the exhaust gases, first, from a coal furnace and, second, from the combustion of city gas. If no gas-fired furnace of large size is available, a Junkers calorimeter may be used to furnish the city gas products.

The excess coefficient for each case should be calculated (see pages 192 and 195), and a theoretical reaction written with the determined amount of excess air. A knowledge of, or assumption upon, the fuel analysis is, of course, necessary. From these reactions, the theoretical products may be expressed in percentages and checked against the analyses.

The student should also determine the following, expressed in pounds per pound of coal or cubic feet of fuel gas:

Weight of dry products of combustion
      of H_2O in the products
      of air supplied
      of carbon incompletely burned (if any),

using the test data on exhaust gases and assuming fuel analyses. For the coal combustion, if there is no knowledge of the quantity of carbon lost through grate and in cleaning, it may be assumed that the carbon gasified equals the total carbon in the coal.

The heat quantities constituting "losses," calculable from the weight quantities, are discussed under Tests 57, 60 and 61.

PART THREE

THE TESTING OF POWER PLANT UNITS

SECTION I

PRIME MOVERS

A prime mover is defined as any machine which converts energy, from a natural source, into useful power. The source of natural energy may be the chemical energy stored in fuels or the energy of falling water. It is evident, then, that prime movers comprise such things as steam engines, steam turbines, internal combustion engines, gas turbines, hydraulic turbines and water wheels.

47. The Determination of Cylinder Clearance

Principles. Linear clearance of a piston engine may be defined as the least distance between the piston and cylinder head in a direction parallel to the center line of the cylinder, when the engine is on dead center. It may have different values at the head end and crank end of the stroke.

Volumetric clearance is the cylinder volume between the piston and nearer cylinder end, which may be occupied by the working medium when the engine is on dead center. It may have different values on the two ends. The volumetric clearance is not only that in the bore of the cylinder, but includes the volume of the ports up to the valve face and the volume of any fittings, such as indicator piping, which may be filled with the working medium in the usual operation of the engine. The space in cylinder drain pipes open to the cylinder is sometimes considered as clearance volume; in some cases, however, this space, in usual operation, is filled with water of condensation so that it is not part of the clearance.

Volumetric clearance is expressed as a part or per cent of the piston displacement.

"Piston displacement" is the volume swept through by one stroke of the piston, that is, the area of the piston times the length of stroke.

A knowledge of the clearance of engines is necessary to an analysis of their performance and losses, especially in connection with indicator diagrams. (See Tests 50(a) and (b).)

(a) **Linear and Volumetric Clearance by Linear Measurements.** The most accurate way to determine linear clearance, if the engine is not too large, is to loosen the connecting rod when the piston is at the end of its stroke, and then note how far the piston rod moves when it is pushed from the dead center position to contact with the cylinder cover. If the engine is too large for this, the cylinder cover may be removed, and the measurement made by compressing, between it and the piston, a small quantity of putty. The putty should be stuck to the cover at the place where the distance between it and the piston, or the piston rod extension, appears to be least. The piston should first be oiled and then dusted with graphite, so that the putty will not stick to it. The putty is then compressed by bolting the cover in place. The least thickness of the putty is the linear clearance. It may be measured by the use of a piece of fine, stiff wire.

The volumetric clearance may be obtained by mensuration from the drawings of the engine. This gives only an approximate result since the actual castings may differ materially from their drawings, and since the position of the piston in the cylinder may change with wear of the connecting rod bearings. If mensuration is used, it is best to make drawings of the engine clearances from the engine as it stands, as nearly as possible to scale.

(b) **Volumetric Clearance by Water Measurement.** The engine should be put on the desired dead center by placing the crank and connecting rod in line by eye. The steam valve (or valves) should be made tight either by smearing its face with heavy cylinder oil or, with slide valves, by squeezing a piece of rubber gasket between the valve face and seat. The piston and cylinder bore should be well oiled to prevent leakage. If the cylinder has holes tapped on the top for indicator piping, they may be used for the introduction of water by which the clearance volume may be completely filled. By taking the weight of the water before and after filling, the amount necessary to fill the clearance space is determined. Dividing this by the density of the water gives the clearance volume. The density may be taken as 0.0361 lb. per cu. in. at 60° F.

If the temperature is materially different from this, it should be noted and the corresponding density used.

Should there be a slight amount of leakage which cannot be prevented, it may be corrected for by determining the "leakage rate." This is done by measuring the amount of water necessary to keep the clearance space full for a period of 1 min. Now, this is probably the maximum rate of leakage, since, during the filling, the leakage probably varied with the head of water in the cylinder. The average leakage may be taken as one-half that shown by the test. The period of time necessary for the filling should therefore be multiplied by one-half the maximum leakage rate to get the total leakage that occurred during filling. This weight subtracted from the weight used in the filling gives the weight corresponding to the actual clearance volume.

It should be noted that there are several assumptions in the foregoing which make the method approximate; it should not be used when the leakage is very large.

If the cylinder to be tested contains pockets in which air may be trapped during the filling, as is likely to be the case with internal combustion engines, these pockets should be blanked off and measured separately if possible. If this is not possible, the water method should not be used.

For accurate results, several determinations should be made and none accepted unless they show fair agreement.

To express the clearance volume in percentage, the piston displacement must be known. The bore may be measured with inside calipers, and the stroke by measuring the distance between dead center positions of the cross-head, a mark being made on the cross-head guides at each end to indicate these positions.

(c) Volumetric Clearance from the Indicator Diagram. Professor Paul Clayton proposed a method based on the fact that gases and vapors expanding or being compressed in closed cylinders follow the law that

$$PV^n = constant$$

This law is represented by a logarithmic curve. Consequently if the amounts of the pressures and volumes obeying it are plotted on logarithmic coordinating paper (the coordinates being proportional to the logarithms of their markings as with the scale of a slide rule), the result is a straight line. In an indicator diagram, the volumes represented are the total volumes of the working medium behind the piston, which include that of the clearance space. In order to transfer the indicator diagram

to logarithmic coordinates, it is necessary to know the clearance volume so that the axis of abscissas may be located. If the indicator diagram is located on logarithmic coordinates by assuming a value of the clearance volume, the resulting expansion and compression curves will be concave to the origin, if the assumed value is too small, and convex if too large. This provides a method of determining the clearance volume, since it is only necessary to plot the expansion or compression curve from the indicator diagram on the logarithmic scale using a number of assumed values of the clearance volume until that one is found which produces a straight line. A convenient procedure is as follows:

Divide the indicator diagram with equidistant vertical lines into ten spaces, and mark the points of intersection of the verticals with the expansion curve 1, 2, 3, etc. The coordinates of these points will be used to locate the logarithmic diagram. The absolute pressures corresponding to these points should be measured from the zero pressure line which is laid off 14.7 lb. below the atmosphere line. After choosing a suitable scale with which to represent these pressures on the logarithmic chart, their positions are indicated by placing small marks against the vertical axis of logarithmic coordinates. Now, let a value of the clearance volume, x, be assumed in per cent. Then the point, 1, corresponds to a volume of 10 plus x per cent; point 2, to a volume of 20 plus x per cent, and so on. Using these volumes expressed as percentages, the points are readily located on the logarithmic chart. If the resulting curve is concave to the origin, a larger value of the clearance is assumed, and so on until that value which produces a straight line is found.

It will be noted that there are limiting values of the assumed clearance between which the straight-line condition seems to be satisfied. On this account the method is recommended only where the clearance is large, as in internal combustion engines; the percentage of error then being not excessive.

48. Valve Setting of a Simple Slide Valve Engine

Principles. It is assumed that the student is acquainted with the principles and operation of the simple slide valve and linkage and with the various quantities pertaining, such as steam lap, exhaust lap, angle of advance, etc., which subjects are amply covered by various works on the steam engine.

With a steam engine, in order to set the slide valve so as to better the distribution of steam, there are only two parts that may be adjusted

without altering the design of the parts, namely, the valve stem, which may be changed in length, and the eccentric, which may be changed in its position on the shaft relative to the crank. The former adjustment affects the laps and lead of the valve; the latter, the angle of advance. A study of the valve motion will show the effects of such adjustments upon the valve and upon the steam distribution as follows:

| | Effect of Increasing | | | |
| | Angular Advance | | Stem Length | |
On	Head End	Crank End	Head End	Crank End
Outside lap.........	Same	Same	Increases	Decreases
Inside lap..........	Same	Same	Decreases	Increases
Lead...............	Increases	Increases	Decreases	Increases
Admission..........	Earlier	Earlier	Later	Earlier
Cut-off.............	Earlier	Earlier	Earlier	Later
Release............	Earlier	Earlier	Earlier	Later
Compression........	Earlier	Earlier	Later	Earlier

The student should confirm this table by drawing the linkage as in Fig. 94 to represent the positions at the various events of the stroke, taking them first for one end of the cylinder and then the other, and observing from the drawings the effects of the adjustments as tabulated. In Fig. 94 the engine is shown at crank end release. The valve stem is broken so as to show the valve on top of the cylinder, for convenience, instead of at the side.
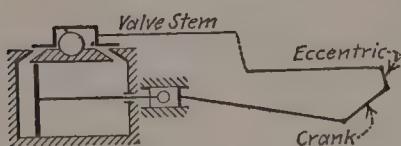
FIG. 94.

It should be noted that the laps may be measured when the eccentric is vertical, as this position of the eccentric puts the valve practically in its mid-position.

(a) **Measurement of Lead.** This is made by measuring the amount of port opening, the valve chest cover being removed, when the engine is on dead center, that is, when the crank and connecting rod are in line. The engine should not be put on dead center by noting the extreme position of the cross-head because, at the end of the stroke, there is no

appreciable movement of the cross-head when the crank moves through several degrees. The eccentric, however, is then in a position where a few degrees of error will cause a considerable error in the position of the valve. Consequently, the dead center must be accurately located. The following method may be used (see Fig. 95).

The crank is turned until it is about 30° from the dead center position sought. A mark, 1, is then made on the cross-head against a mark, 2, on the guides. Another mark, 3, is made on the flywheel against a mark, 4, on some stationary object. Now the engine is turned through its dead center position until the cross-head mark 1 again comes into the coincidence with 2. The crank and connecting rod will then occupy the positions shown by the dotted lines and the mark 3 will be at 5. A new mark is now made on the fly-wheel opposite 4, and the distance between this and 5 on the flywheel rim is bisected by the line 6. If the flywheel is then turned until 6 is opposite 4, the engine will be on dead center. Care should be taken that the motion of the linkage, when advancing to the positions for marking, should always be in the direction of the dead center sought so as to avoid error through lost motion of the parts. If there is no place conveniently near on which to put the mark 4, a pair of trammels should be used.

FIG. 95.

(b) **Setting the Valve for Equal Leads.** From the table given under "principles," it is seen a change in the angle of advance will make the leads on both ends larger or both smaller, while a change of the stem length will make the lead on one end larger and on the other smaller. It will therefore be convenient to change the stem length to *equalize* the leads, and then to adjust the angle of advance until they become the desired *amount*. The procedure is as follows:

The eccentric is first put in its approximate position, that is, about 135° ahead of the crank. The leads are then measured according to (*a*). If the lead on the head end is greater than on the crank end, the stem length must be increased to equalize them, and vice versa (see table). If the leads are then both too small, the eccentric is shifted on the shaft further away from the crank until the desired leads are obtained, and vice versa if the leads are too large.

The proper amount of lead depends upon the travel of the valve, the engine speed, and the amount of compression. More lead is required for high-speed engines, and less when the compression is high. For low-

speed engines, leads up to $\frac{1}{16}$ in. are usual, and for high speeds, twice that amount.
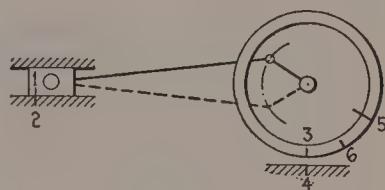
It is advisable, after setting for equal leads, to indicate the engine as a check upon the setting. The lead should be such as to give a vertical line at the admission for both head and crank ends.

(c) Setting for Equal Cut-Offs by the Bilgram Diagram. The typical Bilgram diagram is shown by Fig. 96, the positions of the crank corresponding to the various events of the stroke being shown (for the head
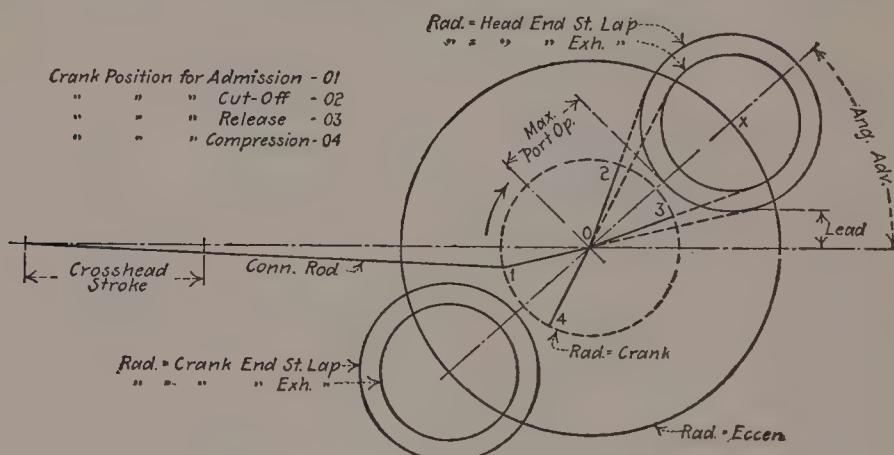
Fig. 96.—Bilgram Valve Diagram.

end only) by the radial lines. The events of the stroke for the crank end may be found by reference to the crank end lap circles with similar construction. For convenience the lower half of the diagram may be revolved through 180° so as to be superposed on the upper half. For other details and for proof, see Halsey's *Valve Gears*.

To lay out the diagram for the purpose of valve setting, we must assume a cut-off, and measure the connecting rod and crank lengths and eccentricity from the engine. If the cut-off is C inches from the end of the stroke, the corresponding crank positions, 01 and 02, may be found by drawing the engine linkage to scale as in Fig. 97. The diagram may be constructed on these lines, starting by putting in the eccentric circle as shown.

The values of the steam laps will depend upon the setting of the valve, but we may locate the lap circles by determining the sum of the steam laps. This equals the length of the valve in the direction of its motion minus the distance between the outside edges of the ports, in the case of the D-valve taking steam outside. The steam lap circles

may be constructed by first locating a point x on the eccentric circle such that the sum of its perpendicular distances from the lines 01 and 02 equals the sum of the steam laps. The lap circles may now be drawn about x as a center and tangent to the lines 01 and 02. The leads and maximum port openings are then determined as shown. If the leads are too large, a later cut-off must be assumed and the construction repeated.

Using the data for the lead and port opening at one end (preferably the crank end, since there they are larger), the valve may now be set
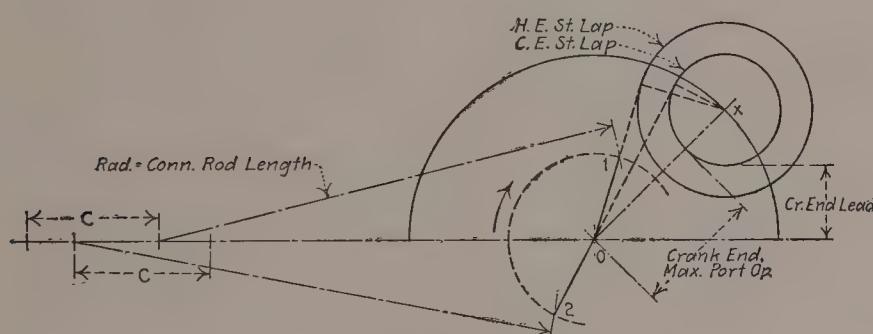
FIG. 97.—Setting for Equal Cut-offs.

for the assumed cut-offs. To do this, the crank is revolved until the valve has moved as far as it will toward the head end of the cylinder, thus giving the maximum port opening on the crank end and making the eccentricity parallel to the engine center line. The stem length is then adjusted until the crank end port is open an amount equal to that indicated by the diagram. Next the engine is put on its crank end dead center by the method given under (a) and the lead made equal to that shown by the diagram by turning the eccentric on the shaft. This completes the setting, but the lead and port opening should be measured on the head end as a check, and compared with the values given by the diagram.

(d) **Valve Setting by the Indicator.** The ultimate test by which the steam distribution must be judged is by the indicator. The valve setting should be such that not only does the engine run smoothly, but the maximum power should be received for the steam used.

Valves may be set by deductions from the indicator diagrams, and the previously described methods dispensed with. The procedure may be more laborious and less systematic, since it is largely cut and try.

Fig. 98 shows indicator diagrams, superposed on ideal ones represented

by dotted lines, from which may be noted the variations caused by too early and too late occurrence of the events of the stroke.

If the admission is too early (that is, if there is too much lead), full boiler pressure is admitted to the cylinder before the beginning of the stroke and the admission line rises at a point other than the extreme of the diagram. If the admission is too late (too little lead), the admission line slants inward because the full boiler pressure does not reach
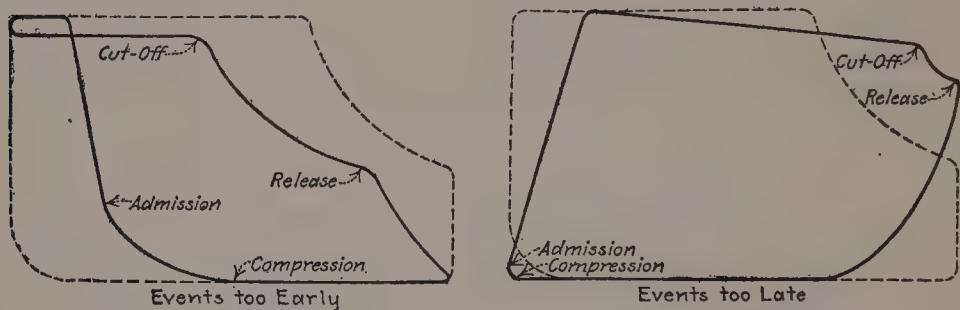

FIG. 98.—Faulty Valve Setting Shown by Indicator Diagrams.

the cylinder until the piston has advanced appreciably in its forward stroke.

The cut-offs may be too early on either end of the cylinder only in relation to that on the other end; they should be approximately equal for equal division of the load. The simple slide valve engine does not cut off earlier than six-tenths of the stroke; if the cut-off on either end is much later than this, as shown by the point at which the expansion line begins on the diagram, either the valve is not well designed or its setting is imperfect.


FIG. 99.

If release takes place too early, the pressure is released from the cylinder before the end of the stroke; the result is a curtailing of the expansion line and a sharp drop down to the back-pressure line. If the release is too late, expansion may take place clear to the end of the stroke, but when the piston returns there is not enough opening for the steam to exhaust through freely. The result is excessive back pressure at the beginning of the return stroke as shown by the rounded toe of the diagram in Fig. 98.

The compressions on the two ends should begin at approximately the same percentage of the stroke. If there is too little compression, the
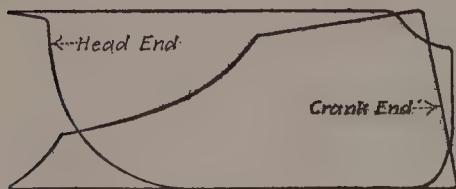
engine will not run smoothly; if too much, the maximum power is reduced, although the economy in the use of steam may be increased. Compression tends to reduce the steam loss due to clearance.

To set a slide valve by the indicator, it is necessary to have diagrams from both ends of the cylinder. Faulty steam distribution may then be determined in detail by the effects on the diagrams as just described.
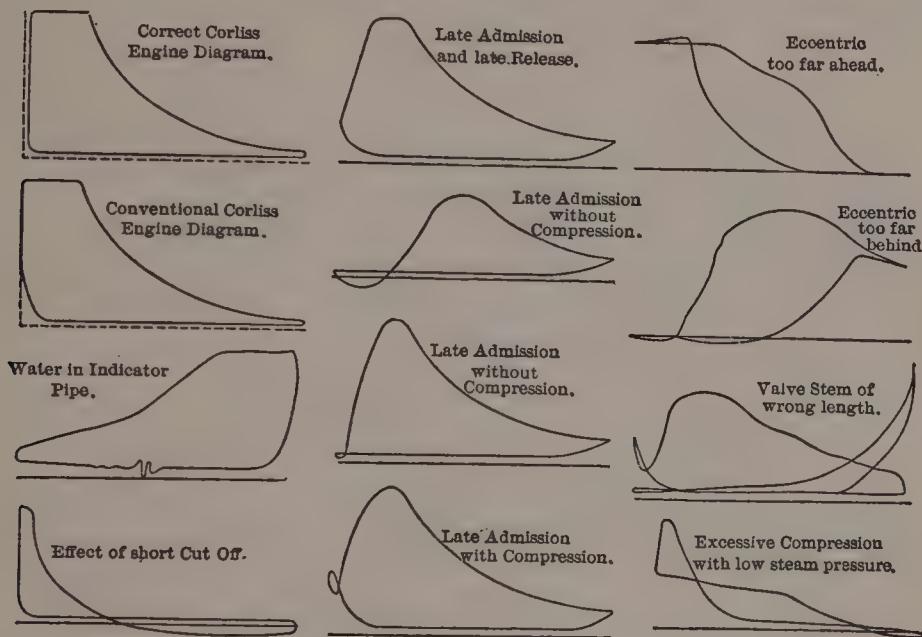

Fig. 100.—Faulty Valve Setting and Diagrams.
(Reproduced from *Power*.)

A knowledge of the effects of the adjustments of stem length and eccentric position (see table under Test 48, Principles), then enables one to correct the steam distribution. The engine should then be indicated again, and further corrections made if necessary. As an example of the reasoning involved, consider Fig. 99. It is well first to make a table showing the existing characteristics of the diagrams, as follows:

	Admission	Cut-Off	Release	Compression
Head end.............	Early	Late	Late	Early
Crank end............	Late	Early	Early	Late

By reference to the table on page 210 it is seen that to change the angular advance will aggravate some of the errors, whichever way it is adjusted. But if the stem length is increased, the events will tend to equalize and change to proper amounts. The stem length should there-

fore be increased and diagrams taken, the procedure being repeated until the cut-offs are about equal. It may then develop that all of the events are too early or too late, in which case the eccentric should be adjusted until the admission lines are vertical. If the release and compression are then satisfactory, the setting is acceptable. Samples of indicator diagrams showing faulty valve setting, etc., are shown by Fig. 100.

49. Setting a Corliss Valve Gear

Principles. The action of the Corliss valve gear is explained in numerous works on the steam engine. The student should have an understanding of this action to grasp the following:

The setting may be considered in two parts: first, for the proper laps and leads, and, second, for the governor regulation. Usual values of the laps and leads are as follows, the smaller ones corresponding to the smaller engine sizes.

$$\text{Steam lap} \dots\dots\dots\dots\dots \tfrac{1}{32} \text{ to } \tfrac{1}{2} \text{ in.}$$
$$\text{Exhaust lap} \dots\dots\dots\dots \tfrac{1}{16} \text{ to } \tfrac{3}{16} \text{ in.}$$
$$\text{Lead} \dots\dots\dots\dots\dots\dots \tfrac{1}{32} \text{ to } \tfrac{1}{8} \text{ in.}$$

The laps are adjusted and measured when the linkage is in its midposition, as indicated by Fig. 101. This figure also names the terms by which the various parts will be referred to, so that the following instructions may be understood.

(a) Adjustment of Laps and Loads. The first step is to set the wrist plate in its center position, the hook rod being lifted to free the wrist plate. There is generally a mark on the wrist plate hub which registers with three stationary marks, as shown in Fig. 101. The middle stationary mark indicates the correct center position of the wrist plate, and the outer marks, its extreme positions. If desired, the center position may be checked with a plumb line. Having located this, the hook rod length should be adjusted so that, when engaged with the wrist plate, the rocker arm stands vertical as shown by a plumb line. The eccentric rod length may now be adjusted so that the eccentric throws the wrist plate to the correct extreme positions.

Now, with the wrist plate in its center position and the steam valves hooked up, the laps may be made the desired amounts by altering the lengths of the steam and exhaust links. The laps are readily measured when the exhaust bonnets are removed, as lines will be found to indicate the edges of the valves and ports. This accomplished, put the engine on

head end dead center (see Test 48(a) for method) and make the head end lead the selected value by turning the eccentric on the shaft, thus moving the valve to the desired position for lead. Then fasten the eccentric, put the engine on the other dead center, and measure the crank end lead. If the leads are not equal, the crank and steam link may be changed in length slightly to obtain equality.

(b) **Adjustment of Governor Rods.** With the governor in the lowest running plane (but not in the safety stop plane), turn the flywheel by
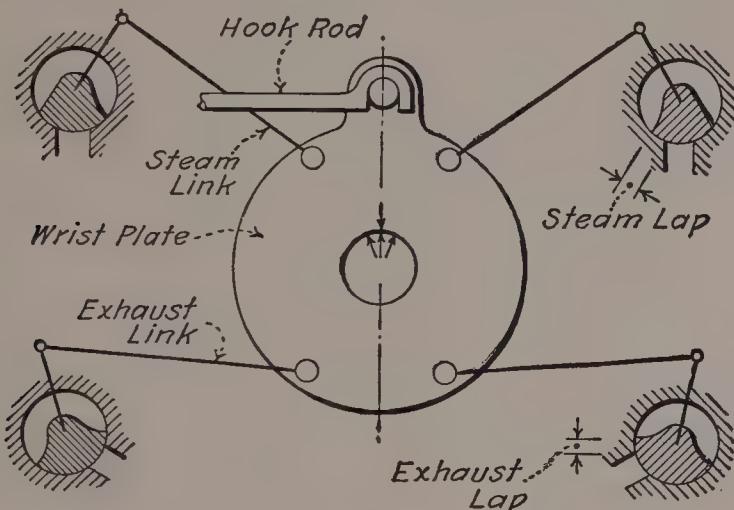

Fig. 101.

hand until the wrist plate is at its limit of travel toward the head end as shown by the mark on the wrist plate hub. The crank-end governor rod having been previously lengthened, the crank-end valve will not trip during the motion. This rod is now slowly and carefully shortened until tripping occurs, the wrist plate being stationary at the head end limit during the adjustment. This done, the head end rod is dealt with similarly. The cut-offs will then be latest when the governor is in its lowest running plane.

The no-load action of the governor should next be checked. Block up the governor to its highest plane, in which position the steam valves should just fail to hook up.

Another procedure is to set the governor rods at the highest position of the governor so as to open the valves a very small amount when the valves are tripped; for the lowest position the valves are not released at all. Note that a setting at one limit of the governor travel cannot be made without affecting that at the other.

(c) **Adjustment of Dash-Pot Rod.** Each steam valve hook engages with a "catch-block" on an arm rigidly fastened to the valve, and this arm is connected with the dash-pot rod. By lengthening or shortening the dash-pot rod, the catch-block may be raised or lowered. When the valve hook is in its lowest position (that is, when the wrist plate is at an extreme position), the corresponding dash-pot rod should be changed at length until there is an equal clearance above and below the catch-block, between it and the hook.

(d) **A Check of the Setting by Indicator** should be made after the completion of all adjustments. In particular, the amount of compression should be ascertained, and the action of the dash-pots toward a sharp cut-off, together with the equalization of cut-offs. Improvements in the setting may be made often by changing the steam and exhaust links.

50. The Mechanical Efficiency Test of a Steam Engine

Principles. The mechanical efficiency of a steam engine is equal to the useful horsepower divided by the horsepower developed by the steam in the cylinder. If a Prony brake is used to measure the useful horsepower, then (see Test 6)

$$\text{B.hp.} = BWN$$

is the useful horsepower.

Using the following notation,

$\qquad$ I.hp. = Indicated horsepower;

P_h and P_c = the mean effective pressure on the head end and crank end, respectively;

$\qquad L$ = the length of the stroke in feet;

$\qquad a$ = the area of the cylinder, in square inches;

$\qquad a'$ = the area of piston rod, in square inches;

$\qquad N$ = the number of revolutions per minute;

then the cylinder horsepower may be expressed,

$$\text{On the head end, I.hp.}_h = \frac{P_h LaN}{33{,}000} = K_h P_h N \quad \cdots \quad (1)$$

$$\text{On the crank end, I.hp.}_c = \frac{P_c L(a - a')N}{33{,}000} = K_c P_c N \quad \cdots \quad (2)$$

K_h and K_c are called the "engine constants."

$$\text{Total I.hp.} = \text{I.hp.}_h + \text{I.hp.}_c$$

Mechanical friction causes a loss of power so that all the work developed in the cylinder does not appear at the flywheel. The friction horsepower equals

$$\text{F.hp.} = \text{I.hp.}_h - \text{B.hp.} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

When the brake is entirely removed from the engine, the work done in the cylinder to keep the engine running is expended to overcome friction only.

The function of the engine governor is to keep the speed constant. This is done by the action of any change of speed, through centrifugal force, to alter the position of the governor weight. More or less steam is admitted into the cylinder according to the position of this weight, and therefore more or less cylinder work is done. To accommodate itself to a variable load, then, the engine *must* allow some change of speed to regulate the cylinder work. At no load, with the governor admitting the least steam, the speed is greatest, and at maximum load, least. Good regulation requires that the difference between these limits of speed be small.

For comparative purposes the speed regulation is expressed as a percentage variation from the mean speed. If N_m and N_n are the speeds at maximum and no load, respectively, then

$$\text{Per cent speed regulation} = \frac{(N_n - N_m) \div 2}{(N_n + N_m) \div 2} \times 100.$$

(a) The Indicated Horsepower is determined by measuring the quantities in equations (1) and (2), the engine constants being calculated before the test. The mean effective pressures are found from indicator diagrams by integration with a planimeter. The speed is determined with a hand or continuous counter.

The brake horsepower being the independent variable, first calculate the net forces that should be obtained at the brake scales to produce the required horsepowers. The engine is then operated at one of these horsepowers by adjusting the tension of the brake strap to give the necessary force at the scales, and to maintain it at that value. This means constant observation of the brake scales and occasional regulation of the brake-strap tension. The engine should be operated at each load for 10 min. or more so that it may adjust itself to the new conditions of load and friction. A number of indicator diagrams and measurements of the r.p.m. are then taken. It is convenient to average the readings

of P_h, P_c, and N at each load and to use the averages for substitution in equations (1) and (2).

A more convenient, although approximate, formula for calculating the indicated horsepower is as follows:

$$\text{I.hp.} = k(P_h + P_c)N$$

in which k is the average of the two engine constants or $L\left(a - \dfrac{a'}{2}\right) \div 33,000.$

A brief consideration will show that the more nearly the area on the crank end is equal to that on the head (that is, the smaller the area of the piston rod is relative to that of the cylinder), the less error there is in the approximate formula. Also, the less difference there is between the mean effective pressures on the two sides of the piston, the less error there is. When they are equal the results from the two methods of calculation are the same.

The following is a rational formula by which the per cent of error ensuing from the approximate formula may be determined for any conditions. Let

$R =$ ratio of P_c to P_h;
$d =$ diameter of the piston rod, inches;
$D =$ diameter of the cylinder bore, inches.

Then

$$\text{Per cent of error} = 50 \times \frac{d^2}{D^2} \times \frac{1 - R}{1 + R}.$$

Using this formula, it will be found, with usual proportions of d and D, that the mean effective pressure on one end must be two or three times that on the other to produce more than 1 per cent of error. It is thus clear that the approximate calculation may almost always be used, since any engine with properly set valve would give a more uniform division of the load than this under normal conditions. Some engines divide the cylinder work quite unequally at light loads; in such a case the formula will show whether or not the correct method should be used. A convenient rule is as follows:

If the mean effective pressure on one end is not greater than

$$\frac{50d^2 + D^2}{50d^2 - D^2},$$

times the mean effective pressure on the other, then the error is less than 1 per cent.

(b) The friction horsepower at each load is found as indicated by equation (3). Tests have shown that this quantity is very nearly constant at all loads of a given engine, if properly operated. As the conditions of lubrication affect friction and therefore mechanical efficiency, the lubrication should be given special attention and maintained uniform throughout the test. It is advisable to plot a curve of F.hp. vs. B.hp. as the test progresses because this is closely indicative of the accuracy of the results.

(c) The efficiency is found by dividing each value of the B.hp. by the corresponding value of the I.hp.

(d) Speed Regulation. Values of N_m and N_n have been obtained for the other results, and these may be used to determine the speed regulation. Another test consists in quickly throwing the entire load on or off and noting the resulting momentary variation in speed. This is greater than that produced by a gradual change of the load, and it should be determined with a chronograph or tachograph.

Any change in boiler pressure will affect the speed regulation, so this should be kept constant during the test if possible, and any variations noted.

51. Economy Test of a Steam Engine

Principles. Economy tests on steam engines are made to determine the amount of steam and heat they consume per unit of power, under different conditions. The most important variable in such tests is the brake horsepower; it is usual to vary it throughout the working range for a complete economy test.

Economy results should always be based on the brake horsepower, but the indicated horsepower is often used because it is inconvenient or impossible to brake the engine. The unit "horsepower-hour" will be used, meaning the amount of work developed in 1 hr. by 1 hp. The heat equivalent of this work should be remembered. Since 33,000 ft.-lb. of work are developed in 1 min. by 1 hp. and 778 ft.-lb. equal 1 B.t.u., then in 1 hr.

$$\frac{33,000}{778} \times 60 = 2545 \text{ B.t.u.} = 1 \text{ hp.-hr.}$$

The steam supplied to the engine is generally expressed in pounds per horsepower-hour, that is, the total pounds of steam supplied in 1 hr. divided by the horsepower. The result should be considered in connection with the condition of the steam, since less will be needed at high pressure or superheat, and more if it is wet. It is therefore necessary to

include a statement of the condition of the steam in the expression for steam consumption.

The heat energy consumed by the engine may be calculated from the total weight of steam supplied and the enthalpy change occurring in each pound. Each pound of steam received by the engine has a certain enthalpy composed of the enthalpy of the liquid, all or part of the enthalpy of vaporization (depending on whether the vapor is saturated or wet). If the steam is superheated there will also be the enthalpy of superheat to make up the total enthalpy.

A part of the enthalpy change, which occurs in the engine, is converted to useful work, another part is lost to friction, radiation, etc., in the cylinder; the balance is rejected in the exhaust. During passage through the cylinder, part of each pound of steam is condensed, so that when it appears in the exhaust, it is partly water and partly vapor at a considerably reduced pressure. The enthalpy of such a mixture is $H_f + xH_{fg}$ so that the heat energy consumed by the engine might be judged to be the enthalpy change represented by the difference between the initial enthalpy and the final enthalpy of the exhaust. That is, the energy consumed is the difference between the heat supplied and the heat rejected.

Of the heat rejected, however, the enthalpy of vaporization is not available without bringing in auxiliary apparatus such as a feed-water heater. Therefore, it is reasonable to charge the enthalpy of vaporization of the exhaust steam against the engine. The enthalpy of the liquid of the exhaust may be reclaimed by being returned to the boiler. Hence, the enthalpy change charged to the engine is:

$$H_{g1} - H_{f2}$$

where H_{g1} is the initial enthalpy of the steam supplied to the engine and H_{f2} is the enthalpy of the liquid of the steam at the exhaust condition. Under ideal conditions the enthalpy of the liquid of the exhaust might be returned to the boiler without loss of heat. That this is not accomplished under real conditions is not considered a fault of the engine.

If S is the weight of steam supplied per horsepower-hour, including moisture if wet, H_{g1} is the enthalpy of the steam near the throttle valve of the engine, and H_{f2} is the enthalpy of the liquid at exhaust conditions; then, the heat energy consumed by the engine per horsepower-hour is

$$S(H_{g1} - H_{f2}).$$

For future purposes it is well to note here that *the heat energy converted into useful work per pound of steam* is

$$\frac{2545}{S}$$

and that S can be based upon either the brake horsepower or the indicated horsepower.

(a) **Steam Consumption.** Having measured the horsepower, it is only necessary to determine the hourly rate of steam supplied. This may be done in various ways as follows:

By Indicator Diagram. The engine itself is a crude form of steam meter, since at every stroke a definite volume of steam is taken into the cylinder which can be measured from the indicator diagram. Referring to Fig. 60, at point b cut-off takes place, the cylinder is closed to the boiler, and the cycle commences with a volume of steam equal to that represented by point b. This comprises the clearance volume and the volume of the piston displacement up to point b, or $(c + C)D$ cubic feet; c being the clearance expressed as a part of the piston displacement; C the cut-off expressed as part of the stroke; and D the piston displacement in cubic feet. Assuming the steam having this volume to be saturated, we may find its density in pounds per cubic foot from its pressure. Calling the density W, we have $W(c + C)D$ pounds as the weight of the steam at point b. Not all of this steam has been furnished by the boiler at the beginning of the cycle, however, since some steam from the previous stroke was compressed in the clearance space. Just before the valve opened to the boiler, the steam in the cylinder had a pressure and volume corresponding to point e. Its weight then is wcD, in which w is the density of saturated steam at the pressure at e. The amount of steam furnished by the boiler to one end each revolution is therefore $W(c + C)D - wcD$ pounds. If N is the revolutions per minute, the weight per hour is

$$60ND[W(c + C) - wc].$$

This is the weight used on one end of the cylinder on the assumption that the steam is saturated just after cut-off. The weight on the other end may be found similarly. Now, the steam is *not* dry because of initial condensation, and it may have varying degrees of wetness, depending upon the cut-off, the working range of temperatures, type of engine, etc. Numerous empirical formulas have been proposed for the calculation of cylinder condensation, an important item, since for simple engines it amounts to between 20 and 50 per cent of the amount of steam shown by the diagram.

The clearance should be determined as described under Test 47. The expression for cut-off, C, may be found by dividing the distance of b, Fig. 60, from ae, by the length of the diagram, in inches.

For the sampling of diagrams, see Test 10(e).

By Condenser. All of the steam used is passed into a surface condenser, and the condensate weighed for a counted time. This is probably the most accurate way of testing for steam consumption. *Numerous time-quantity readings should be made to determine uniformity of flow.* The condenser must be operated with more circulating water than is used in practice, so that the condensate will emerge cool; otherwise a large amount may escape unweighed by evaporation. The piping between the condenser and engine must be examined for tightness, and the condenser tested for leakage. The latter may be done by running the air-pump when no steam is flowing, and noting if any circulating water is drawn through, preferably when the condenser is hot. If it leaks a small amount, a leakage rate may be determined and applied as a correction to the results. If the engine is to be tested "noncondensing," that is, with no vacuum, the condenser must be vented in order to establish this condition.

By Feed-Water Measurement. For this method, the engine and boiler supplying it should be isolated so that all of the steam generated by the boiler is used in the engine. Sometimes it is necessary to supply steam to auxiliary apparatus, such as a feed pump, from the boiler supplying the engine tested. In such a case, it is necessary to make a separate measurement of this steam, either by establishing the rate beforehand or, better, by condensing the steam used by the auxiliary during the test. It is necessary to examine the boiler and piping for tightness, the latter especially at branches stopped by valves. This is done by closing all valves in branches and the main stop valve at the engine so that the supply pipe is open from the boiler to the engine valve, but closed everywhere else. With a quiet furnace fire so that there is no active evaporation, the level is then noted in the water column at a number of uniform intervals of time. If the level falls, leakage is taking place, and the rate should be determined from the area of water surface calculated from the measurements of the boiler. This leakage may then be applied to the results of a test as a correction.

The feed water may be measured by any of the methods given under Economy Test of a Steam Boiler.

By Steam Meter. Since the flow of steam to a reciprocating engine is of a pulsating character, the commercial forms of steam meters are not

suitable for measuring the steam consumption of a reciprocating steam engine.

Willan's Law states that the weight of steam per unit of time used by an engine with a throttling governor varies directly as the indicated horsepower of the engine, very nearly. The work represented by an indicator diagram in which the expansion follows the law that $PV = a$ constant (very nearly the case with steam) is mathematically proportional to the initial pressure, cut-off being constant. As the pressure is proportional to the density of the steam, approximately, it follows that the weight of the steam is proportional to the work, and the indicated horsepower. The weight of steam used also varies directly with the brake horsepower, since B.hp. = I.hp. — a constant. Experiment has shown that this relation applies not only to throttling engines, but to those of the cutoff type, and to steam turbines.

The practical application of Willan's law lies in the consequence that if the weight of steam used per hour by an engine is plotted against its indicated or brake horsepower, the result is a straight line. A slight exception to this appears in the case of engines of the Uniflow type. In such engines the Willan's line is likely to exhibit a slight concavity upward but the deviation from a straight line is small. This furnishes a check upon the results of a test as the test proceeds; if points so plotted do not follow a straight line, there is error. It should be noted that at overloads, the curve is likely to deviate slightly from straightness, and lean toward the axis of steam weights.

The duration of the test should depend upon the uniformity of conditions, the capacity of the engine, and the method used for measurement of the steam. During each test at a given load, weighings are made at uniform time intervals, say, 10 or 15 min. When there are four to six of these of nearly equal amounts recorded consecutively, the run may be discontinued, provided the error of starting and stopping is within reasonable limits. (See Rules for Testing.) This error depends upon the method of measuring the steam. If a condenser is used, the error will be relatively small, since it equals the difference in the amounts of condensate in the condenser at starting and stopping. If the boiler method is used, the test must be much longer, as there may be large error in the level of the water column, due to differences of density, ebullition, etc. With a steam meter, the test need be only long enough to get sufficient indicator diagrams for a fair average, provided the engine has been previously given a settling run, but the accuracy is questionable.

Determinations of the condition of the steam supplied as to quality

and pressure, and of the pressure or temperature of the exhaust, should be made and included in the statement of steam consumption per horse-power-hour.

The relation between the steam consumption in pounds per indicated horsepower-hour, S_1, and per brake horsepower-hour, S_2, is

$$S_1 = S_2 \times \text{mechanical efficiency.}$$

(b) Cylinder condensation is often figured by subtracting the total steam used per hour, as shown by the indicator diagram, from that determined by direct measurement. The difference includes not only cylinder condensation, but valve leakage in the case of engines which do not have separate valves controlling the exhaust.

(c) Thermal Efficiency. There are two standards, one having a commercial, the other a scientific, value. The former is the ratio of the heat equivalent of the useful work to the heat units consumed as defined under "principles." Since there are $S(H_{g1} - H_{f2})$ heat units consumed for every horsepower-hour of useful work, and since the heat equivalent of a horsepower-hour is 2545 B.t.u.,

$$E = \text{thermal efficiency} = \frac{2545}{S(H_{g1} - H_{f2})}$$

in which S may be based on indicated or brake horsepower. If the former, the result is the "cylinder efficiency"; if the latter, it is the "overall efficiency."

H_g may be obtained from the pressure and quality determinations of the steam supplied, by use of the steam tables; H_f is determined similarly, from readings of a pressure gage or thermometer in the exhaust.
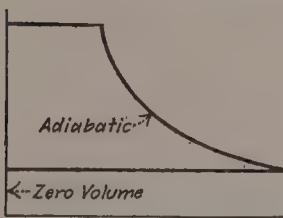

FIG. 102.—Rankine Cycle.

Efficiency Ratio. This is the ratio of the actual engine efficiency (as defined) to that of an ideal engine working without clearance space and with non-heat-conducting cylinder walls with cut-off early enough to allow expansion clear down to the back pressure. There will then be no clearance loss and the expansion will be adiabatic. The indicator diagram representing these conditions is shown by Fig. 102, and is known as the Rankine cycle, although it is by some writers attributed to Clausius. The initial pressure and quality of the steam and the back pressure are assumed to be the same in the ideal engine as in the actual. The "efficiency ratio" is a more reasonable standard for efficiency than the

commercial standard since the latter charges against the engine heat that it could not use under even ideal circumstances because of the limitations in the operation of the working medium.

The ideal *cycle* efficiency is the heat available for work, per pound of H_2O, divided by the heat added per pound. Under the conditions of the assumed ideal engine the heat converted into work is the difference between the total heat H_{g1} of the steam supplied and the total heat of the exhaust H_g', since there are no losses. H_g' is the theoretical amount of heat left in a pound of steam after it has expanded adiabatically from the given pressure and condition as to quality to the given back pressure. It may be calculated by the use of entropy tables, but it is more convenient to determine it directly from the Mollier total heat-entropy diagram which gives the heat of steam under all conditions and at various stages of adiabatic expansion.

Since the heat added per pound is the same in the ideal cycle as that for the actual engine as previously defined, the *cycle* efficiency is

$$(H_{g1} - H_g') \div (H_{g1} - H_{f2}).$$

Then the

Rankine Eff. ratio $= E_c = \dfrac{2545}{S(H_{g1} - H_{f2})} \div \dfrac{H_{g1} - H_g'}{H_{g1} - H_{f2}} = \dfrac{2545}{S(H_{g1} - H_g')}.$

Since $2454 \div (H - H')$ is the steam consumption, S_c, in pounds per horsepower-hour of the ideal engine

$$E_c = \frac{2545}{H_{g1} - H_g'} \times \frac{1}{S} = \frac{S_c}{S}.$$

This is a useful form, since by it the cylinder or over-all efficiency may be readily obtained, depending upon the basis of S.

52. Test of a Multiple Expansion Engine

Principles. The results to be sought are in part identical to those for a simple engine; hence the principles under Tests 50 and 51 are appropriate and should be read in this connection. In addition, there are other data useful to the study of multiple expansion, principally pertaining to the indicator diagrams. The **sampling** of diagrams is therefore very important, and should be done according to Test 10(*e*).

Although, for brevity, a compound engine only will be considered here, the methods apply equally to any multiple expansion.

(a) **I.hp., B.hp., F.hp., and mechanical efficiency** may be found as for a simple engine, but in many cases the unit will be too large to brake conveniently, or a generator connection will make that procedure impossible.

In the former case, the F.hp. may be determined by running the engine free and taking indicator diagrams, called under these circumstances "friction diagrams." The I.hp. from such diagrams equals the friction horsepower. If the F.hp. is assumed constant at all loads, this single determination of it at zero load enables the calculation of the B.hp. at any load, since

$$\text{B.hp.} = \text{I.hp.} - \text{F.hp.}$$

When there is a direct connected generator, the electrical load should be measured as for a steam turbine (Test 53(a)). If the efficiency curve of the generator is known, the B.hp. of the engine may then be estimated quite accurately.

The I.hp. may be obtained by figuring that for each cylinder separately, and adding them to get the total I.hp. Or the "equivalent mean effective pressure" (to be defined later) may be used in a single calculation upon one cylinder.

Whenever possible, the B.hp. should be used as the independent variable (if results from the engine alone are to be considered).

(b) **Steam consumption** by condenser or feed-water measurement. (See Test 51(a).)

(c) **Thermal Efficiency.** (See Test 51(c).)

(d) **Equivalent and Aggregate M.e.p.** The equivalent M.e.p. referred to any cylinder may be defined as a pressure of such value as to produce the same horsepower in the referred to as in the actual cylinder. Thus,

Let A_h and A_l = net piston areas for high- and low-pressure cylinders, respectively.

P_h and P_l = the M.e.p.'s high- and low-pressure cylinders, respectively.

Then the M.e.p. of the high-pressure cylinder, referred to the low, is

$$\frac{A_h}{A_l} \times P_h,$$

and the M.e.p. of the low-pressure cylinder, referred to the high, is

$$\frac{A_l}{A_h} \times P_l.$$

The combined or "aggregate" M.e.p. is one of such value that, if it prevailed in the cylinder referred to, there would be produced in that cylinder an I.hp. equal to that actually produced by all cylinders. That is (calling the aggregate M.e.p., P_{hl}),

$$\text{Referred to the H.P. cylinder, } P_{hl} = P_h + \frac{A_l}{A_h} \times P_l.$$

$$\text{Referred to the L.P. cylinder, } P_{lh} = P_h \times \frac{A_l}{A_h} + P_l,$$

and similarly for three or more cylinders.

(e) **Steam Accounted for by Indicator Diagrams.** The weight of steam per hour shown by an indicator diagram equals *

$$60ND[W(c + C) - wc]$$

Now, for a single cylinder engine, in which displacement is D cu. ft.,

$$\text{I.hp.} = \frac{PLaN}{33,000} = \frac{PN(D144)}{33,000}.$$

Dividing the one equation by the other, we have

$$\text{Wt. of steam per hp.-hr.} = \frac{13,750}{P}[W(c + C) - wc],$$

in which P is the mean effective pressure, c and C are the clearance and cutoff volumes expressed as parts of the piston displacement, respectively; and W and w the densities of saturated steam at the pressures of cutoff and compression, respectively. This last equation is the more convenient form for the diagram water rate.

The procedure then is to measure C, W, and w on representative diagrams (c being known) from the cylinder considered. The aggregate M.e.p. referred to that cylinder (defined under (d)) is then calculated and taken as the value of P in the water rate formula. The result is the steam accounted for by the diagram from the cylinder considered. The procedure is repeated for the other cylinders.

(f) **Combined Diagram.** Let

D_h and D_l = piston displacements, cubic feet, of high and low-pressure cylinders, respectively.

* This formula is sometimes quoted with $w(c + k)$ in place of wc, $c + k$ then standing for the volume of the steam at the beginning of the compression curve.

v_h and v_l = clearance volumes, cubic feet, of high- and low-pressure cylinders, respectively.

Select representative diagrams from the cylinders, as in Figs. 103 and 104, and divide them into ten or more equal spaces by vertical lines.

Choose convenient scales of pressure and volume for the combined diagram, Fig. 105, and lay them off on coordinate paper. Draw in the atmospheric pressure line.
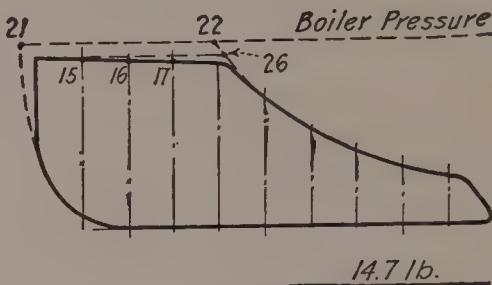
FIG. 103.—H.P. Diagram.
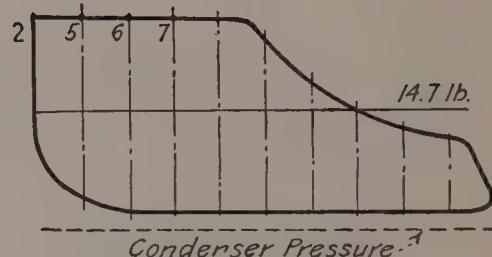
FIG. 104.—L.P. Diagram.
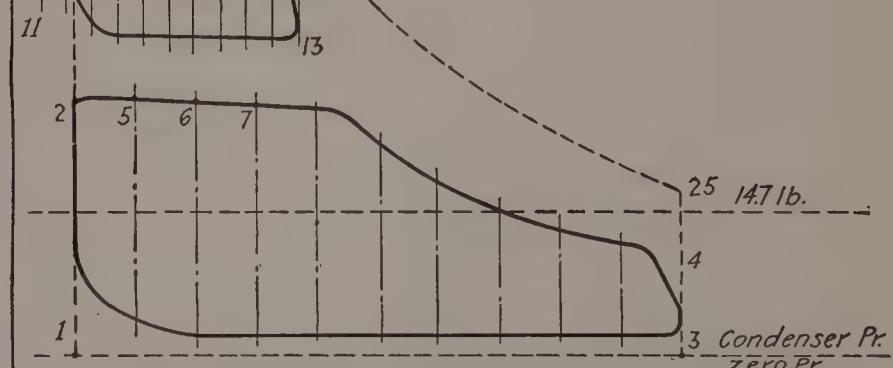
FIG. 105.—Combined Diagrams from a Compound Engine.

Locate the vertical line 1–2, Fig. 105, at the volume graduation corresponding to the clearance volume, v_l. Locate the vertical 3–4 at the volume graduation corresponding to $v_l + D_l$.

Make ten (or more, corresponding to Fig. 105) spaces between lines 1–2 and 3–4 with equidistant vertical lines.

Points 5, 6, 7, etc., on the reconstructed diagram may now be located on these verticals by scaling the pressures of the corresponding points of Fig. 105 from the atmospheric line. In this way the full low-pressure diagram may be reproduced on the combined diagram.

Locate vertical 11–12, Fig. 105, to represent the clearance volume v_h; and 13–14 to represent $v_h + D_h$.

Draw equidistant verticals between 11–12 and 13–14, and locate points 15, 16, 17, etc., as for the low-pressure diagram. This establishes the high-pressure diagram.

The combined diagram, Fig. 105, fairly represents the whole expansion of the steam on a single P–V scale. When making comparisons with assumed ideal expansion diagrams, however, it should be borne in mind that Fig. 105 does not truly represent the continuous expansion of a given weight of steam. The high-pressure expansion curve is for all of the steam in the high-pressure cylinder; but the low-pressure expansion is for a lesser quantity of steam by the amount caught in the clearance of the high-pressure cylinder. Furthermore, valve and piston leakage will affect the validity of comparisons made with the hyperbolic expansion curve or the constant steam weight curves.

To draw in the ideal expansion line for the given conditions (the ideal being assumed hyperbolic, or $PV = $ a constant), the boiler pressure line should first be drawn in on both the high-pressure and combined diagrams, as indicated on Figs. 103 and 105. The compression line of the high-pressure diagram (Fig. 103) is continued on a hyperbolic curve until it intersects the boiler pressure line at point 21. The expansion curve is continued similarly, thus locating point 22. The volume corresponding to the distance between 21 and 22 is now calculated in cubic feet. This volume is assumed to be that of the steam received from the boiler per stroke, as though in an engine without clearance.

Now continue line 1–2, Fig. 105, until it intercepts the boiler pressure line at 23. From 23 lay off 23–24 equal to the volume just calculated. Through 24, construct the required hyperbolic curve by one of the well-known methods, using the line 1–2 and the condenser pressure line as zero axes.

(g) **The ratio of expansion** is equal to the volume of the low-pressure cylinder, including clearance, divided by the volume of the steam in the high-pressure cylinder, including clearance, at the point of cutoff. The last named is found as follows and is then termed the "commercial cutoff." Referring to the high-pressure diagram, Fig. 103, a horizontal is drawn through the highest point on the steam admission line. The intersection of this horizontal with the prolonged expansion curve, that is point 26, is the commercial cutoff.

(h) **The diagram factor** may be found by dividing the area of the combined diagram, Fig. 105, by the area of the ideal diagram, 23–24–25–4–3–1. Or it may be calculated by dividing the aggregate M.e.p. referred to the low-pressure cylinder by the ideal M.e.p. obtained from the following formula:

$$\text{Ideal M.e.p.} = \frac{P'}{R'} (1 + \log_e R') - p,$$

in which P' = boiler pressure, pound per square inch, absolute;

p = condenser pressure, pound per square inch, absolute;

$$R' = \text{ideal ratio of expansion} = \frac{\text{length 1–3}}{\text{length 23–24}}.$$

53. Economy Test of a Steam Turbine

Principles. The measurements and results, in general, are the same as those for a reciprocating engine, except that there can be no indicated horsepower determination, and often no brake measurements, an electrical load being considered instead. Since it is not possible to indicate a turbine, the term "internal horsepower" is sometimes used to express the equivalent of indicated horsepower, but it is not a definite quantity. It is often impracticable to apply a brake to the turbine shaft on account of the direct connection of a generator; in such a case the turbine and generator must be tested as a single unit. For other details, see Test 51, Principles.

(a) **Determination of the useful horsepower** may be made the same as for Test 50 if the turbine alone is tested with the use of a brake. Otherwise, the electrical load should be measured in kilowatts by taking either wattmeter readings or readings of amperes and volts. The horsepower equivalent to the electrical output, that is, the "electrical horsepower," may be found from

1 Electrical horsepower (E.hp.) = 0.746 kw.

For alternating current generators, the A.S.M.E. Test Code for Steam Turbines states that the net electrical output is equal to the electrical output of the generator minus that portion of the excitation power that is separately supplied minus power separately supplied for purposes of ventilation. The excitation power is the product of the current, supplied to the generator field, and the sum of the voltage drops across the generator field and the main field rheostat. Only that portion of the excitation power which is separately supplied from a source external to the engine-or-turbo-generator unit is to be charged against the prime mover.

(b) **Steam Consumption.** The pounds of steam consumed may be measured by the following methods:

1. For a turbine exhausting to a surface condenser:
   By weighing or measuring the condensate.
2. For a turbine exhausting to atmosphere, or to a jet condenser:
   (a) By weighing or measuring the feed water to the boiler which is supplying steam to the turbine only, or
   (b) By removing and calibrating the first stage nozzle.

Dividing the total steam, per hour, measured by one of the above methods, by the horsepower or kilowatts output gives the pounds of steam per horsepower or kilowatt-hour.

It is often desirable, for purposes of comparison, to base the steam consumption on brake horsepower or "internal horsepower." The former quantity may be estimated, if an efficiency curve of the generator is available, by multiplying the steam consumption in pounds per electrical horsepower-hour at a given load by the generator efficiency at that load.

The internal horsepower is difficult to estimate with any degree of accuracy. It is sometimes assumed to be the quotient of the brake horsepower and the mechanical efficiency of a steam engine working under the same conditions. This, of course, is a crude estimate. Using it, however, the steam consumption may be expressed on the basis of internal horsepower.

(c) **Duration of Test.** Each of the constant load runs in an economy test should continue for not less than 1 hr., when the condensate is measured or weighed. In case the boiler feed water is being measured or weighed, the constant load runs should continue for not less than 10 hr. in order to eliminate the inherent errors in that method of measurement.

Each constant load run should be preceded by a preliminary period sufficient to establish constancy of conditions.

(d) Additional Data. If there are traps arranged to catch condensation from the turbine casing, they should be drained regularly and the condensate weighed in with the steam consumed. Readings should be made of the pressure in the nozzle chamber to show the drop of pressure through the governor valve. If this drop is excessive, the steam consumption will be correspondingly high. The predetermined conditions of pressure, quality, and back pressure or vacuum should be kept as constant as possible, and the quantities carefully measured, as they have a decided influence upon the economy of a turbine. It is important that the barometer be read to get the required accuracy in the measurement of vacuum which should be expressed as an absolute pressure. This is because the heat content of steam at low pressures varies materially with the pressure, and also because turbine economy is more dependent upon the predetermined vacuum than pressure or superheat.

For the purpose of estimating the steam consumption of a turbine under a given set of conditions of steam pressure, quality and vacuum, such as are stated in the manufacturer's guarantee, the turbine being tested under somewhat different conditions, correction curves are supplied by the manufacturer. These show the number of pounds of steam per horsepower- or kilowatt-hour to be subtracted from the test result if the stated pressure is higher than the actual, the number to be subtracted if the stated superheat is higher, and the number to be subtracted if the stated back pressure is lower, or vice versa.

Willan's line should be plotted during all trials under variable output.

(e) The thermal efficiencies are obtained exactly as for a reciprocating steam engine, the basis for the computation of useful work being the electrical, brake, or internal horsepower.

(f) Separation of Losses. The losses of a turbine may be considered in two groups. First, approximately constant losses which include those due to friction and radiation. Frictional resistance is encountered at the bearings and stuffing boxes, and between the discs and blades and the steam, and through windage of the external parts. Second, steam losses including leakage through clearance spaces, condensation, and faulty action of the blades and nozzles in not completely absorbing the energy of the steam.

These losses may not be separately measured without an involved test. Professor Carpenter has indicated a method of separating the two groups

by the use of Willan's line. When plotted against brake horsepowers, this line intercepts the axis of steam consumption at a point which shows the steam consumed per hour to overcome the first group of losses, that is, at no load. The distance parallel to the load axis, measured in horsepower, necessary to move Willan's line so that it shall pass through the origin, equals the power given to the first group of losses. At any other load, this quantity is to be subtracted from the power that would be developed by the actual amount of steam consumed if operating on the Rankine cycle; the result is the actual brake horsepower plus the second group of losses, from which the second group may be readily obtained.

54. Test of a Hydraulic Turbine

Principles. The horsepower delivered by a hydraulic turbine is measured by a dynamometer applied to its shaft. The horsepower available is that of the water which drives it, and is proportional to the difference in level between the head-water surface and the tail-water surface, and the weight of water flowing per minute.

Measurement of Head. The *gross* or *total hydrostatic head* is obtained by measuring the difference of level between the head-water surface and the tail-water surface when the plant is not operating. The measurement of the *net* or *effective head* depends somewhat on the type of plant. For an *open-flume turbine* the effective head is the difference in level between the head water in the flume, near the center of the turbine, and the tail water at the end of the draft tube. In some cases the velocity head of the head water coming into the turbine flume is regarded as a part of the effective head. For an *encased turbine* the effective head is the sum of the pressure head, the velocity head at the casing intake and the difference in level between the point of measurement and the end of the draft tube. For an *impulse turbine*, without a draft tube, the effective head is considered to be the pressure head at the nozzle plus the velocity head at the nozzle plus the difference in level between the nozzle and the tail-water surface.

Power. The *potential energy* of any hydraulic plant is energy latent in the available flow of water due to the difference in level between the head-water surface and the tail-water surface. The horsepower thus available is called the *theoretical horsepower* (t.hp.) and is expressed as follows:

$$\text{t.hp.} = \frac{Q \times W \times H}{33{,}000}$$

where Q = cubic feet per minute
W = weight per cubic foot
H = total hydrostatic head.

The *water horsepower* is that resulting from the product of the weight of water flowing by the effective head.

$$\text{w.hp.} = \frac{Q \times W \times H'}{33{,}000}$$

where Q = cubic feet per minute
W = weight per cubic foot
H' = the effective head as defined above.

Efficiency. There are two efficiencies which are used in connection with the tests of hydraulic turbines: (1) the plant efficiency and (2) the turbine efficiency. The plant efficiency is the quotient of the brake or output horsepower divided by the theoretical horsepower. The turbine efficiency is the quotient of the brake horsepower divided by the water horsepower.

$$\text{Plant eff.} = \frac{\text{B.hp.}}{\text{t.hp.}}$$

$$\text{Turbine eff.} = \frac{\text{B.hp.}}{\text{w.hp.}}.$$

Rate of Flow. The rate of flow of the water supplied to the turbine should be measured either by direct discharge methods or by the velocity-area method. The former requires the weighing or measurement of volume of the water, flowing in a given period of time, in calibrated tanks. The velocity area method requires the determination of the cross-section of the stream flowing in the conduit and the mean velocity of the stream. A number of methods are available for this purpose such as Pitot tube, venturi meters, and weirs. There is still another method known as the *salt velocity* method which consists of introducing a salt solution at a known rate into the stream and determining the amount of salt in the stream water at a point far enough downstream to insure thorough mixture of the salt solution with the water. From these data can be calculated the quantity of water flowing.

Selection of the Independent Variable. In operation, any or all of the following may be varied. First, gate or needle valve opening; second,

rotative speed; and third, brake horsepower. For test purposes, the head on the turbine may also be varied.

Change of the gate opening varies the amount of water supplied. Change of the brake horsepower is accompanied by a change of speed if the gate opening is left the same.

Laboratory tests are usually made at a constant head. The gate is adjusted at a predetermined opening for one series of tests. Then, by regulating the brake, the speed is varied for this series, and data obtained at each speed. Another series of tests is then made at a different gate opening, and so on until the full range has been covered.

If it is desired to test the performance at different heads, all or part of the above tests may be repeated after changing the head.

It is useful to plot brake horsepower and efficiency at each gate opening against speed.

Duration. The accuracy of an efficiency test of a hydraulic turbine depends mainly on the method employed for measuring the flow of water through the turbine. A sufficient time must be allowed to obtain accurate measurements of the rate of flow when the head remains constant after uniform conditions have been established.

Determination of Best Operating Speed. The curve of efficiency versus speed is something like an inverted U. The speed at the highest point of this curve gives maximum efficiency. It is approximately that speed which gives a peripheral velocity of one-half that of the jet for impulse wheels. For other types, it depends upon the characteristics of the turbines.

Under ideal conditions, the water upon leaving the turbine vanes would have no velocity except that due to the acceleration of gravity, since velocity of the off-flow means lost energy. With some types of turbine the best speed may be ascertained by watching the off-flow and determining that speed which is accompanied by the most nearly vertical descent of the water as observed by the eye.

55. Adjustment of an Internal Combustion Engine

Principles. Modern internal combustion engines work on either the Otto or Diesel cycle. The following discussion will be limited to the Otto cycle but many of the principles are common to both. For an extended discussion on the adjustment of Diesel engines, see any standard work on the subject.

By far the greater number of present-day internal combustion engines operate on the Otto cycle which requires four strokes for a complete series of events. First, the suction stroke, in which the piston, advancing from the head end, takes in a charge of gas or vapor fuel mixed with air. Second, the compression stroke, returning, in which the charge is compressed. Third, the expansion stroke, at the beginning of which the mixture is burned and afterward performs the useful work. Fourth, the exhaust stroke, during which the burned gases are discharged. Fig. 106 shows this cycle of operation on a *PV* diagram. There are two valves to control the charge, one for inlet and one for exhaust. Fig. 107 shows where these valves open and close relative to the crank position and on the indicator diagram. It is seen that the inlet valve opens a little after dead center. This is arranged so that it will open after the exhaust valve has closed; otherwise there would be a tendency for the exhaust gases to enter the intake. The closing of the inlet valve takes place after crank end dead center so as to provide a good opening for the charge. The corresponding crank angle may be as much as 20° to 30°, with high-speed
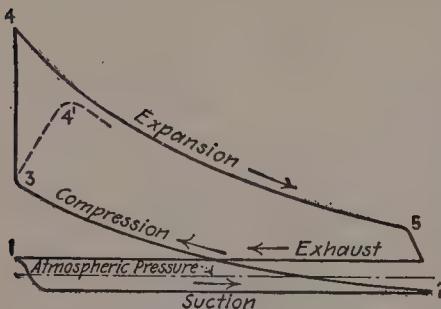
FIG. 106.—Otto Cycle.

Inlet Opens.   Inlet Closes. Exhaust Opens. Exhaust Closes.
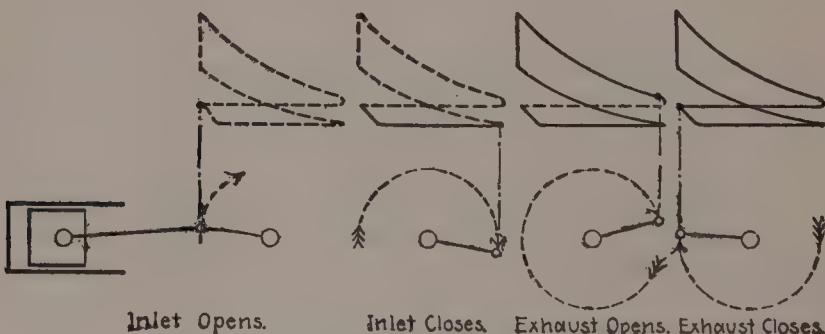
FIG. 107.—Gas Engine Valve Events.

engines; the momentum of the incoming gases causes them to continue to flow into the cylinder during the inward piston motion. The exhaust valve should open a little before crank end dead center and close a little after the opposite dead center to allow free egress of the burned gases, the amount depending upon the speed and type of the engine. The correct angular distances of the crank from the dead center positions, corre-

sponding to the valve events, are usually stated by the engine manu-
facturer.

Ignition of the charge should occur in the neighborhood of dead center;
generally before, when the engine is running. This is because it takes
an appreciable time for the gas mixture to rise to its maximum pressure
after ignition. For maximum power, the greatest pressure should occur
at the beginning of the stroke; consequently, ignition should "advance"
this position. The greater the rotative speed, the greater will be the
angular distance, or advance, between the crank and dead center when
ignition takes place. Some fuels burn more slowly than others; for such
the ignition must be more advanced. The angular dis-
tance varies between zero and 50°.

(a) **Timing the Valves.** This is done by reference to
the crank positions when the valves are opening or clos-
ing. A prick point should be made on the flywheel rim
to mark the line of the crank, and the position of this
prick point located by trammels or otherwise when the
engine is on dead center. The angular distance between
the prick point and its dead center position will deter-
mine the motion of the crank for any valve event.

Fig. 108.
Gas Engine Valve.

The valves usually are operated by cams on a cam
shaft driven by 2 to 1 gears, a and b, from the main
shaft (see Fig. 108). There is a certain amount of clearance between the
valve stem and the cam when the latter is not acting. By changing this
clearance the timing of the valve will be altered. From the figure it will
be seen that to make the clearance greater makes the valve open later
and close earlier. The timing may be changed also by changing the
relative positions of the gears a and b. Thus, in the figure, if gear a is
turned back (clockwise) so that it meshes with one tooth behind on b,
both opening and closing will be later.

There is usually some provision made for changing the clearance. The
procedure in timing the valves, then, is first to locate the crank positions
corresponding to opening and closing, and then to correct, if necessary,
by changing the clearance or gear mesh or both. The point of opening
or closing may be accurately fixed by turning the valve stem with the
fingers; the friction of the valve when seated is easily felt. It should be
noted that the valve events are somewhat different when running because
of the expansion of the valve stems due to the working temperature. To
allow for this, the engine may be operated until it warms up, and then

timed. The makers' instructions for timing are usually for the cold condition.

With high-speed engines, such as are used in automobile and marine practice, it is customary to set the valves by making each clearance a stated amount, and this amount is usually very small, between 0.002 and 0.006 in. The purpose is to minimize the noise and wear which are produced by excessive clearance. Clearances are often made on such engines as small as the mechanic can make them. Systematic procedure calls for the use of "spacer" gages, or "feelers," by the use of which the clearances may be made exact.

The adjustment of the clearance is preferably made after the engine has been run long enough to be brought to operating temperature. In certain types of engines, notably the "vee-type," this is not possible and the clearances must be adjusted with the engine cold. In this case, the clearances must be larger to allow for expansion of the metal when heated.

When adjustment of the clearance has been made, the timing of the valves should be adjusted. Good timing of the exhaust valves is more important than good timing of the inlet valves. Most modern engines are built with only one camshaft and with the cams an integral part of the shaft. Nothing can be done about the relative position of the inlet and exhaust cams but the proper relation of the camshaft to the crankshaft can be adjusted.

(b) Timing the Ignition. Practically all modern engines use the high-tension magneto or a jump-spark system employing a storage battery, as source of power, and a high-tension coil in conjunction with breaker-points and a distributor. The best method of adjusting the timing of the ignition is to use a spark protractor.

This device consists of a disc, attached to the crankshaft, which carries a small neon bulb in a small pocket in the disc. A narrow slit is cut from the pocket to the rim of the disc so that the neon light appears as a narrow band parallel to the axis of the crankshaft. The neon light is fired but the high-tension ignition system of one of the cylinders and the bulb must be located so that it is at the top of its rotation when the piston, of the particular cylinder, is at top dead center at the beginning of its power stroke.

Beside the rotating disc is a stationary protractor scale, graduated in degrees of arc, with the zero at the point corresponding to top dead center. The position of the spark advance can readily be seen by noting the position of the neon flash with respect to the graduated scale.

For economical operation, ignition should be advanced as far as possible without causing knocking, "pinging," or diminution of speed. Some engines are equipped with automatic spark-advance devices which retard the spark for starting and then advance it when the engines attain a certain speed.

(c) **Adjustment of Mixture.** Whether the engine uses oil vapor or fuel gas, there is one device for setting the amount of air supplied and another, for the amount of fuel. Strictly speaking, these should be readjusted for every load on the engine, but usually a single adjustment is made to give the best mixture at usual running loads. As a general rule, it may be accepted that the most economical mixture is as dilute a one as may be had without causing misfiring or objectionable slowing down of the engine. This effect is due to the lower working temperatures of the expanding medium. Also, for economy, the fuel valve should be open as little as possible. This is especially true with engines having throttling governors. If the engine is given too much fuel, it tends to speed up, putting the work of throttling upon the governor. Thus, the suction in the cylinder is greater and the line 1–2, Fig. 106, is lower. Thus the greater the throttling, the larger will be the lower loop of the indicator diagram. This represents the work lost in pumping the charge into the cylinder.

With an engine regulated by a governor, the procedure is as follows: The desired load is applied by brake or otherwise and the valve controlling the flow of fuel is set with a slightly larger opening than is thought to be correct. The air opening is then increased until the engine shows signs of slowing down, as determined by a tachometer. Records are kept of speed and valve settings for this trial. The procedure is then repeated with a slightly less fuel valve opening. The smallest fuel valve opening with the largest air valve opening which will not cause objectionable slowing down or irregular firing may be accepted as the best setting for the applied load. In this connection it is essential that the exhaust be observed during each trial; it should be clear and regular.

(d) **Adjustment of carburetors** for variable speed engines supplied with gasoline, such as automobile and marine engines, is more difficult than the adjustment of mixing valves on constant speed engines, but the same principles are to be observed. Many carburetors are made with automatic adjustment to accommodate changes of speed, and the adjustment is made with the engine delivering no torque. This procedure is convenient, but not productive of best results. In general, it is a good plan to follow the manufacturer's instructions for carburetor adjustment

to get what may be considered an **approximate** setting. A readjustment then should be made, *under a load* representing the average running condition. This readjustment involves reducing the needle valve opening and increasing the auxiliary air opening to a point just short of back firing. This will give the most *economical*, although not the most *powerful*, mixture. Also, it will probably be necessary to determine another setting for starting the engine when cold.

Finally, the action of the carburetor should be judged by the appearance and sound of the exhaust.

(e) Test of Timing by Indicator. This applies to engines running at not more than 250 r.p.m. if the ordinary indicator is to be used. Special gas engine indicators are made to handle about double this speed. Higher than this, only optical or other special high-speed indicators may be used with success. A light spring should be used in the indicator, about 20 lb., and the indicator piston arranged with a stop so that it will not register much above atmospheric pressure, in order to get the suction diagram on a larger scale.

If the inlet valve opens too late, the line from 1, Fig. 106, will not have as sharp a drop as it should and the suction line will be lower. If the inlet valve closes too late, the compression curve will not start sharply from the point 2. If the exhaust valve opens too late, expansion is carried a little beyond point 5, but the beginning of the exhaust line will be higher on account of the greater back pressure. If it closes too late, the effect is much the same as for too late opening of the inlet. When the valve events are too early, opposite effects will be recorded.

There is an important connection between the timing of the exhaust valve and economy in the use of fuel. This has to do with the "scavenging" of the cylinders, that is, the cleaning out of the exhaust gases, which is necessary to the complete combustion of the next charge. Hence the exhaust valve should be kept open for as long a crank motion as other conditions permit.

The indicator diagram does not show fine differences in the timing of the valves because the events occur near the ends of the stroke where the indicator drum has but little motion corresponding to the angular motion of the crank. For this reason, it is useful to set the reducing motion of the indicator 90° ahead of the crank, so that the valve events will be shown in the middle of the indicator diagram.

If the ignition is properly advanced, the line 3–4, Fig. 106, will be vertical. If ignition is too late, this line will slant as shown by the dotted

line 3–4′, causing a diminution of power; if too early it will slant in the opposite direction with the same effect, accompanied by pounding.

56. Mechanical Efficiency Test of an Internal Combustion Engine

Principles. Exactly the same methods and formulas apply to this test as to Test 50, with the exception of the determination of N in the formula for indicated horsepower. Internal combustion engines are generally single acting in moderate sizes, so, with throttling governors and regular firing, there is only one power stroke for each two revolutions of a crank. Under these conditions, the value of N is one-half the revolutions per minute. When the engine governs on the hit-and-miss principle, however, the actual number of explosions per minute must be counted. This also applies with a throttling engine if the firing is irregular.

The indicator diagram, Fig. 106, contains a negative area which must be subtracted from the upper area to determine the net mean effective pressure. This may be done with the planimeter by traversing the upper area in a clockwise direction, and the lower, anticlockwise. The indicated horsepower obtained from the result is the "net" indicated horsepower which is greater than the brake horsepower by the horsepower given to mechanical friction. The lower loop of the diagram gives the work lost to fluid friction, that is, to pumping the charge into the cylinder. It may be determined accurately only by using a light indicator spring as explained under Test 55(e).

(a) Determination of Indicated Horsepower. (See Test 50(a).) If the engine governs on the hit-and-miss principle, the explosions may be counted by timing the sound of the exhaust if the speed is not too fast. Special devices are made for this purpose when a continuous record is wanted. One may be made by tapping a small pipe into the exhaust and covering it with a flap held down by a spring. An explosion causes the flap to rise and so actuate the lever of a revolution counter.

(b) Friction Horsepower. The mechanical friction is obtained as for Test 50(b). The fluid friction may be expressed as horsepower calculated from the mean-effective pressure of the lower loop, the value of N being the same as for the net indicated horsepower.

Another method of measuring the friction horsepower, particularly useful in the case of high-speed engines which cannot be indicated successfully, is to connect the engine to an electric dynamometer and drive the engine. The power required to drive the engine is the friction horsepower. It will be noted that the amount of throttle opening affects the

power required to drive the engine at any given speed. This follows directly from the discussion under principles.

(c) **Mechanical Efficiency.** (See Test 50(c).

(d) **Speed Regulation.** (See Test 50(d).

57. Economy Test of a Gas Engine

Principles. A complete economy test of a gas engine should include, besides the fuel consumption, the determination of the various losses so that their distribution may be studied. All of the heat energy of the fuel supplied must be accounted for in the form of useful work and losses. The equation between these quantities is called the *heat balance*.

The losses involved in the operation of a gas engine are as follows: mechanical friction, heat carried away by jacket water, by the dry exhaust gases, by steam in the exhaust, by unburned fuel gas, and stray losses as radiation, etc. It is convenient and logical to base the calculation of these losses upon 1 cu. ft. of fuel gas under standard conditions of temperature and pressure. The heat balance may then be written, "heat supplied by 1 cu. ft. of gas equals the heat from 1 cu. ft. turned into useful work plus heat from 1 cu. ft. lost to friction plus, etc."

To measure the useful work and friction and jacket losses on this basis it is necessary to meter the fuel supplied in a given time. The other quantities of the heat balance require the analyses of both the fuel and exhaust gases.

The table of Constituents of Gas Fuels in Part II gives the average composition of the principal gas engine fuels.

The exhaust gas analysis may be made with the Orsat apparatus. The chief constituents are CO_2, O_2, and N_2, but an analysis for CO should not be omitted. A determination of unburned hydrogen or hydrocarbons is needed only if the air-fuel ratio is low, or if other exceptional conditions prevail.

Knowing the fuel and exhaust gas analyses, the weights of gases in the exhaust pipe per cubic foot of fuel supplied may be calculated from the relations presented in Part II, Products of Combustion. This division should be re-read and thoroughly understood.

Sampling. For a chemist's analysis of the fuel gas, it is best to take a continuous sample covering the whole time of the test. This can be done by connecting a 5-gal. flask with rubber tubing to the gas main and syphoning water from the flask at a slow and uniform rate thus drawing

in the sample. Sampling for heat value determination may be done as described under Heat Value of Fuels.

For throttling engines, the exhaust gas may be sampled as explained under Products of Combustion. Hit-and-miss engines, it should be remembered, occasionally discharge air into the exhaust pipe which has not mixed with fuel. This air carries a certain amount of heat, although less than that carried by burned gases, and for this reason should be counted separately. It is possible to make an arrangement in the exhaust pipe by which the discharge from a missing stroke is by-passed and its temperature and quantity measured separately from the main exhaust. As this is a difficult matter, results may be obtained by averaging in the air from a missing stroke with the working exhaust, the sample then being taken as for a throttling engine. A thermometer in the exhaust pipe will show a mechanical average of the variable temperature which may be used for calculating the heat lost. When the exhaust from a hit-and-miss engine is sampled in this way, the excess coefficient cannot be figured from the analysis without first applying a correction for the air taken in by the engine, but not used to mix with the fuel.

Since both the fuel and exhaust gases are generally under pressure, it is easier and more accurate to obtain samples than in boiler work.

Duration of Test. This should be several hours, preferably, but, on a small engine, a test only 1 hr. long will give valid results if the engine is previously run for 15 or 20 min. under the conditions to be maintained during the test. The readings to be mentioned later should be taken sufficiently often to obtain a fair average. The useful horsepower should be maintained at a constant value and all other conditions kept as uniform as possible. The test results are not valid unless the gas consumption is uniform as shown by meter readings taken at equal time intervals.

To make the methods clear, the measurements from a gas engine test will be worked through for the various results. Table 1 gives these measurements, and also the notation used in the formulas.

The fuel gas analysis is given in Table 2, and is the same as that used for illustration in the section on Combustion.

(a) Fuel Consumption, standard cubic feet of gas per hour, and per hp.-hr. The relations previously derived for gas combustion are for dry fuel gas at 14.7 lb. per sq. in. and 32° F. It should be noted, however, that the standard of the American Gas Association, upon which heating values are based, refers to a temperature of 60° F., and 100 per cent humid gas. The 32° standard must be used in applying the com-

TABLE 1

GIVING DATA FROM A GAS ENGINE TEST AND NOTATION USED IN FORMULAS

B.hp. = brake horsepower   = 25
I.hp. = net indicated horsepower   = 30.1
F.hp. = friction horsepower
V = number of cubic feet of fuel gas per hour, by meter   = 534
V' = ditto, corrected to absolute pressure of 29.92 in. of mercury and 32° F.
B = Barometer reading, inches of mercury   = 29.70
    pressure of fuel gas above atmosphere, at meter, inches of water   = 2.9
F, F' = fuel consumption, standard cubic feet of gas per brake horsepower-hour, and per I.hp.-hr., respectively
T_f = temperature of fuel gas at meter, degrees F. (dry bulb)   = 82
    temperature, wet bulb, in gas main (relative humidity of fuel gas, 83%)   = 77.6
t_j = temperature of ingoing jacket water, degrees F.   = 61
T_j = temperature of outgoing jacket water, degrees F.   = 133
T_e = temperature of exhaust gases, degrees F.   = 950
t = temperature of air near engine, degrees F.   = 80
W_j = weight of jacket water, pounds per hour   = 1220
V_d = volume of dry exhaust gas from the combustion of 1 cu. ft. of fuel, cubic feet per cubic foot
W_d = weight of dry exhaust gas from the combustion of 1 cu. ft. of fuel, pounds per standard cubic foot
W_v = weight of water vapor, ditto
K = ratio by volume of air consumed to fuel gas
X = excess coefficient
c, h, g = from sums of columns (3), (4), (5), Table 2, respectively.

VOLUME ANALYSIS OF EXHAUST GAS, PER CENTS

D = carbon dioxide, CO_2 = 9.85
O = oxygen, O_2 = 4.50
M = carbon monoxide, CO   = 0.20
H = hydrogen, H_2 Not analyzed.
N = nitrogen, N_2 = 85.45

bustion equations, but the 60°, 100 per cent humid, standard must be used for heat values.

It is often the case that the fuel supplied a gas engine *is not 100 per cent humid*. The heat value test of the gas by Junkers calorimeter is, however, necessarily for 100 per cent humid gas. Allowance should be made according to the principles observed on pages 164 and 177. The equation

$$V' = V \times \frac{492}{T_f} \times \frac{P - P_v}{29.92}$$

TABLE 2

ANALYSIS OF FUEL GAS USED DURING TEST, AND COMPUTATION OF
QUANTITIES USED IN FIGURING RESULTS

(1)	(2)	(3)	(4)	(5)
Constituents of Gas	Per Cent by Volume	Number of Mols, per Hundred Mol-Volumes of Fuel		
		of C	of H$_2$	of O$_2$
CO............	15.5	15.5	....	7.75
H$_2$..........	43.4	....	43.4	
CH$_4$........	22.7	22.7	45.4	
C$_2$H$_4$.......	3.5	7.0	7.0	
O$_2$..........	0.5	....	....	0.50
CO$_2$........	3.3	3.3	....	3.3
N$_2$..........	11.1			
Total, per hundred mol-volumes of fuel......		48.5	95.8	11.55
		mols of C = c	mols of H$_2$ = h	mols of O$_2$ = g

may be used, but it should be noted that P is the absolute pressure in the gas main supplying the fuel for test, and P_v is the partial pressure of the H_2O in this gas, allowing for its measured humidity, both P and P_v being in inches of mercury at 32°.

Applying the data of the test (Table 1), the volume of gas supplied, as metered, is 534 cu. ft. at a total pressure of $29.7 + 2.9 \div 13.6 = 29.91$ in. of mercury. The partial pressure of the H_2O in the gas is $0.83 \times 1.1 = 0.19$ in. of mercury. The number of standard cubic feet supplied per hour is therefore

$$V' = 534 \times \frac{492}{82 + 460} \times \frac{29.91 - 0.91}{29.92} = 470.$$

The fuel consumption based upon the horsepower-hour is

$$F = \frac{470}{25} = 18.8 \text{ standard cu. ft. per B.hp.-hr.}$$

$$F' = \frac{470}{30.1} = 15.6 \text{ standard cu. ft. per I.hp.-hr.}$$

(b) **Heat Supplied.** Since the heat balance is based upon one standard cubic foot of fuel gas, the heat supplied is its heating value as determined by Junkers calorimeter and referred to a dry condition at 32°.

The test data show the heating value to be 496 B.t.u. at 60°, 100 per cent humid. For dry gas at 32°, the heating value is

$$496 \times \frac{520}{492} \times \frac{29.92}{29.4} = 532 \text{ B.t.u. per cu. ft.}$$

29.4 being the partial pressure of the gas, in a gas-steam mixture at a total pressure of 29.92 in., the partial pressure of the steam (at 60°) being 0.52 in.

The gas engine code of the A.S.M.E. specifies that the "higher" heat value be used, that is, the heat obtainable from the fuel including the latent heat of vaporization of the water formed by combustion of the hydrogen. Some authorities contend that the lower heat value is more logical to use in this connection since the gas engine cannot avail itself of this latent heat, but for the sake of uniformity the recommendations of the code will be followed.

(c) **Heat Converted into Useful Work.** The heat equivalent of 1 B.hp. is 2545 B.t.u. Since the fuel consumed to produce this power is F cubic feet, the heat equivalent of the useful power per cubic foot of fuel is $2545 \div F$, or for the test figures,

$$\frac{2545}{18.8} = 136 \text{ B.t.u.}$$

Dividing this by the heat value gives 25.6 per cent. This is the thermal efficiency.

(d) **Heat Lost to Mechanical Friction.** In power units,

$$\text{F.hp.} = \text{I.hp.} - \text{B.hp.}$$

The heat equivalent to this loss is $2545 \times$ F.hp., in B.t.u. per hour. Therefore,

$$\text{Friction loss, B.t.u. per standard cubic foot of fuel} = \frac{2545 \text{ F.hp.}}{V'}.$$

Applying the data of the example, this is

$$\frac{2545(30.1 - 25)}{470} = 27.6 \text{ B.t.u.}$$

or

$$27.6 \div 532 = 5.2\%.$$

(e) **Heat Lost to Jacket Water.** This is the heat absorbed in a given time divided by the fuel used in the same time. Considering hourly quantities,

$$\text{Jacket loss} = \frac{W_j(T_j - t_j)}{V'}.$$

The weight W_j may be measured by a barrel or tank on a platform scales. The temperature should be read with two thermometers; one in the discharge tank (provided the piping is arranged to discharge without material radiation loss), the other in a thermometer well in the inlet pipe or in a receptacle drawing water from a tap in the main situated similarly to the jacket as to temperature.

Applying the data of the example,

$$\text{Jacket loss} = \frac{1220(133 - 61)}{470} = 187 \text{ B.t.u.}$$

or

$$187 \div 532 = 35.2\%.$$

(f) **Volume of Dry Exhaust Gas per Cubic Foot of Fuel Gas.** This quantity is to be used in the various calculations from the exhaust gas analysis. In the formula,

$$V_d = \frac{c}{D + M},$$

c is found as in column (3) of Table 2.

For the date of the example

$$V_d = \frac{48.5}{9.85 + 0.20} = 4.81 \text{ cu. ft.}$$

(g) **Loss of Heat to Dry Exhaust Gases.** This determination involves the measurement of the temperature rise of the fuel and air mixing with it, and of the weight of the dry exhaust gases.

The physical condition of the dry exhaust gas is such that it is hardly worth while to make an accurate determination of its specific heat to take account of variation with temperature and gas composition. The prevailing constituent is nitrogen. The mean specific heats in B.t.u. per pound between 60° and exhaust gas temperature, T_e, of nitrogen and of a mixture containing 90 per cent nitrogen and 10 per cent carbon dioxide follow:

T_f	N_2	Mean Value of C_p for 90% N_2 and 10% CO_2
400	0.246	0.243
800	0.25	0.248
1200	0.252	0.251

A constant value of $C_p = 0.245$ will yield an inappreciable error. On page 194 there is deduced the formula,

$$W_d = \frac{V_d}{9000} [11D + 8O + 7(M + N)],$$

by which the weight of the dry exhaust gas may be found from its analysis.

An accurate determination of the exhaust gas temperature is very difficult. Probably the best method is to insert a thermocouple in the exhaust line. It should be located as near the exhaust valve as possible and should project well into the center of the gas stream. Even then the indicated temperature will be less than the actual temperature because the walls of the exhaust passage are at a temperature much lower than that of the gas and the thermocouple will radiate heat to them. A shielded thermocouple will offset this loss to some extent but will be very difficult to install.

Applying the data of the example,

$$W_d = \frac{4.81[11 \times 9.85 + 8 \times 4.5 + 7 \times (0.2 + 85.45)]}{9000},$$

$$= 0.398 \text{ lb. per cu. ft. of fuel,}$$

and loss to dry exhaust gas $= 0.245 \times 0.398 \times (950 - 80)$

$$= 84.5 \text{ B.t.u.}$$

or

$$84.5 \div 532 = 15.9\%.$$

(h) Loss of Heat to Water Vapor in the Exhaust. Since the higher heat value of the fuel is used, the latent heat of the water vapor is to be included. The small amount of water vapor brought in as humidity of the air and fuel may be ignored, particularly as its latent heat is not added by the fuel. This leaves only the water from the combustion of hydrogen. A formula for the weight of vapor so formed from 1 cu. ft.

of fuel gas is deduced in the section on Products of Combustion. Combining this with the expression for total heat of steam under the given conditions, we have,

Loss of heat to water vapor $= W_v \times$ (total heat of the vapor above room temperature)

$$= 0.0005h \,(1090 + 0.46T_e - t).$$

The value of h is found similarly to c, as shown by column (4), Table 2. Applying the data of the example,

Loss to water vapor $= 0.0005 \times 95.8 \times (1090 + 0.46 \times 950 - 80)$

$$= 69.0 \text{ B.t.u.}$$

or

$$69 \div 532 = 13.0\%.$$

Had the total heat of the vapor been calculated from the more accurate expression for the specific of superheated steam the result would have been 1482 B.t.u. per lb., instead of 1446 B.t.u., by the empirical equation. But the corresponding heat loss per cubic foot of gas is only 2 B.t.u. greater. Therefore the shorter calculation may be used.

(i) **Heat Lost by Unburned Fuel Gas.** The volume of unburned CO per cubic foot of fuel gas is $V_i = V_dM \div 100$. As the heat value of CO is 318 B.t.u. per cu. ft., at 68° F., the

Loss to unburned CO $= 3.18\, V_dM.$

Applying the data of the example,

Loss to unburned fuel $= 3.18 \times 4.81 \times 0.2 = 3.06$ B.t.u.

or

$$3.06 \div 532 = 0.6\%.$$

(j) **Heat lost to radiation, etc.,** is found by subtracting from the heat value of the fuel the other quantities of the heat balance (c) to (j), or by subtracting from 100 per cent these quantities expressed in per cents. The unaccounted for heat includes that used for pumping the fuel mixture into the cylinder if the area of the lower loop has been subtracted from the upper of the indicator diagrams. This quantity could be separately measured, but it is generally sufficient to include it in the friction loss as is done when the lower loop is altogether ignored, or to list it under radiation and "unaccounted for."

Applying the data of the example,

Loss to radiation, etc. $= 532 - 136 - 27.6 - 187 - 84.5 - 69 - 3.06$

$$= 24.8 \text{ B.t.u.},$$

or

$$24.8 \div 532 = 4.7\%.$$

(k) Cubic Feet of Air per Cubic Foot of Fuel Gas. The relation for this, deduced on page 194, is

$$K = [V_d(D + O + 0.5M) + 0.5h - g] \times 0.0478$$

in which g is found under column (5), Table 2.

Applying the data of the example,

$$K = 0.0478[4.81(9.85 + 4.5 + 0.5 \times 0.2) + 0.5 \times 95.8 - 11.55]$$

$$= 5.06 \text{ cu. ft. of air per cu. ft. of fuel gas.}$$

(l) The Excess Coefficient. The theoretical air-fuel ratio is

$$0.0478(c + 0.5h - g)$$

which for the data at hand equals 4.05 cu. ft. Consequently the excess coefficient is

$$5.06 \div 4.05 = 1.25.$$

(m) The Real Mixture Standard. In order that the investigator may know how nearly the efficiency of an engine approaches the maximum value obtainable, the ideal cycle efficiency must be calculated. This should be done by taking into account variable specific heats, dissociation (chemical equilibrium), and actual working medium before and after combustion. The computations involved are somewhat laborious but such an analysis has been made by Goodenough and Baker,* who have computed the maximum possible efficiencies which can be obtained from the Otto and Diesel cycles. The results of their analysis are shown in curve form in Figs. 109 a-d.

A comparison between the indicated thermal efficiencies, obtained from actual test, and those set forth in the curves, for similar conditions, furnishes the investigator with a *true efficiency ratio* which indicates how closely the real engine tested has approached the maximum possible efficiency. While the maximum efficiency can never be attained in practice, the closeness with which the actual efficiencies approach the theoreti-

* University of Illinois, Engineering Experiment Station Bulletin No. 160.

cal efficiencies represents the degree of performance attained. Efficiency ratios based on the *air-standard efficiency* result in values which discredit the performance of the engine.
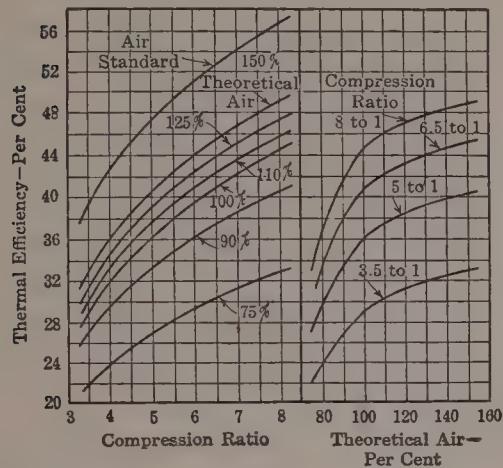
FIG. 109a.—Variations of Efficiency and with Mixture Strength for Otto Cycle.
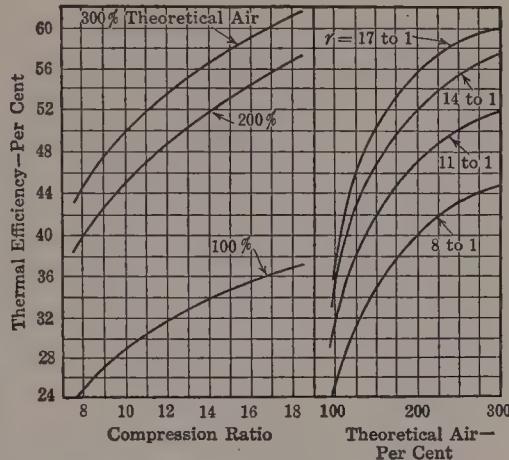
FIG. 109c.—Effect of Compression Ratio and Mixture Strength on Efficiency of Diesel Cycle.
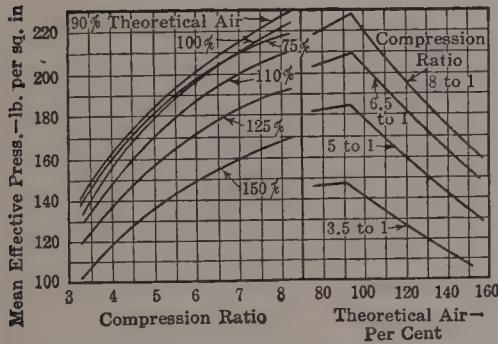
FIG. 109b.—Variation of M.e.p. with Compression Ratio and with Mixture Strength for Otto Cycle.
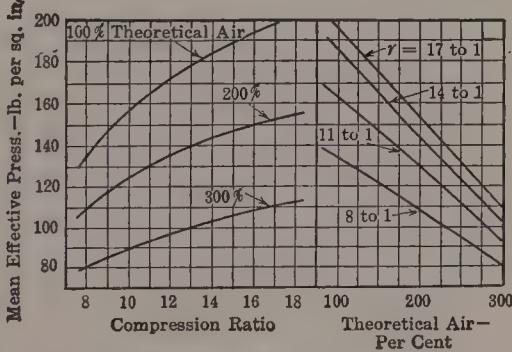
FIG. 109d.—Effect of Compression Ratio and Mixture Strength on M.e.p. of Diesel Cycle.

(After Goodenough and Baker, Eng. Exp. Sta., Univ. of Illinois.)

58. ECONOMY TEST OF A DIESEL ENGINE

Principles. The determinations to be made are, in general, the same as those for an Otto cycle engine. With fuel injection by compressed air, it is desirable to measure, in addition, the power absorbed by the air compressor.

With two-cycle Diesel engines, the power to operate the scavenging air cylinder, or pump, is of importance.

If a supercharger is employed, the power required for this should be measured separately, if possible.

A characteristic of the ideal Diesel cycle, and also of the real engine, is that the thermal efficiency based upon cylinder work increases with decrease of load. With the real engine, the mechanical efficiency (that is, brake horsepower divided by indicated horsepower) decreases with the load. The net effect is that the thermal efficiency based upon the brake horsepower is nearly constant over a wide range of load.

As almost the same amount of air is used per suction stroke at all loads, but the fuel charge decreases with decrease of load, the per cent of CO_2 in the exhaust has very small values at light loads, and increases with increased load. The exhaust gas temperatures range materially lower than those for Otto engines, since a greater percentage of the heating value of the fuel is converted into work.

(a) **Heat Balance.** All of the items listed under Test 57 should be included. The necessary experimental data are exactly parallel, as are also the methods of computation of results. (See Combustion of Oils.) There is, however, an additional item, namely, the energy supplied to injection air. Energy to drive the fuel pump, being small, may be neglected.

(b) **Cylinder vs. Shaft Horsepower.** The difference between gross I.hp. and the B.hp. is the friction horsepower (fluid and mechanical) plus horsepower consumed by injection air compressor, etc. If facilities for measuring the power absorbed by the compressor are lacking, and if suction indicator diagrams are not taken, these items may be expressed in total by a single figure. Great care should be taken to secure accurate indicator records. Because of the form of the pressure-volume diagram of the Diesel cycle, small inaccuracies cause large errors in mean effective pressures.

(c) **Injection Air.** The energy given to the air compressor is not a net loss since some of this energy is returned to the cylinder by the medium of high temperature and high pressure air, capable of increasing the area of the indicator diagram. Consequently the net energy absorbed by the compressor equals the heat lost to its jacket water plus friction generated external to the air cylinders plus heat radiated from air cylinder to atmosphere. Friction developed in the compressor cylinders is converted into heat appearing in either or both jacket water or discharged air.

Neglecting radiation of heat to the atmosphere, an approximation of the energy wasted by the air compressor may be made by assuming it equal to the heat lost to air compressor jacket water, and to any intercooler that may be used. Measurements are then reduced to those for cooling water losses.

(d) **Scavenging Air.** For two-cycle engines, if this air is supplied from separate cylinders, or the crank end of the main cylinder, the mean effective pressure and the indicated horsepower of these cylinders are readily obtained. The friction horsepower can only be reckoned in combination with that of the main cylinder and other moving parts.

If scavenging air is supplied by some form of blower, the power required can only be determined from a separate test of the blower.

It is to be noted that scavenging air does not contribute to the energy developed in the main cylinder, as does injection air.

(e) **Air Standard and Real Mixture Standard.** If the ideal Diesel cycle could be performed with air as a working medium, and if the specific heat of the air remained constant throughout the cycle, then the ideal cycle efficiency, or air standard, can be expressed thus:

$$\text{Efficiency} = 1 - \frac{1}{1.4 r^{0.4}} \times \frac{(c^{0.4} - 1)}{(c - 1)},$$

in which r is the ratio of compression, that is, the ratio of the piston displacement plus clearance volume to clearance volume; and c is the ratio of the volume of the air in the cylinder at the point of "cutoff" (that is, at the beginning of the adiabatic expansion) to the clearance volume.

To apply this to test data, the clearance volume must be determined. At any load, the value, c, may be found from representative indicator diagrams at that load.

As for the Otto cycle, the air standard is unfavorable to the real engine, as a basis of comparison, and for the same reasons. The "real mixture standard" has been calculated for various conditions for Diesel engines, as for Otto engines, and should be referred to in preference to the air standard. (See Test 57(m).)

59. Economy Test of an Automotive Type Engine

Principles. Engines of this type are variable speed. A complete test will therefore consist of enough runs at different speeds to cover the working range. Since the rotative speeds are high, these engines cannot

be indicated except with a special high-speed indicator and at the higher speeds even optical indicators are not satisfactory unless they have been so constructed that inertia effects are practically eliminated.

The exhaust gas analysis cannot always be relied upon for the determination of the losses, because irregular firing, incomplete combustion, or lubricating oil vapor in the exhaust will make erroneous the percentages as usually found. If conditions are such that the exhaust gas analysis is valid, the methods described in the section on Products of Combustion are applicable. In these formulas, $C_g = C_t =$ weight, in pounds, of carbon in the fuel per pound of fuel; since all the carbon is gasified. In the formula for the weight of water vapor in the exhaust, the value of m, moisture in the fuel, drops out. The results from the formulas will be based on one pound of fuel.

The specific gravity of American gasolines varies from 0.71 to 0.76, at 60° F., and contains roughly 83 per cent of carbon, 15 per cent of hydrogen, and less than 1 per cent of oxygen by weight. Kerosene is a little higher in carbon and lower in hydrogen. The lower heating value of gasoline ranges between 18,500 and 19,000 B.t.u. per lb., the average being about 18,800 B.t.u. per lb. The lower heating value of kerosene is about 20,000 B.t.u. per lb. The specific gravity of kerosene ranges between 0.79 and 0.83 at 60° F.

(a) Determination of Torque and Brake Horsepower at Various Speeds. The ordinary Prony brake is very difficult to use when it is sought to maintain a uniform load long enough for a fuel consumption test, because of the lack of uniformity of the frictional resistance. Fan brakes have been used for this purpose, but they are not strictly reliable, nor are water brakes entirely satisfactory due to lack of sensitive control. The best results ensue from the use of an electric dynamometer similar to that described in Part I.

There are several combinations of variables possible, making it necessary to decide upon a definite independent variable. For example, a series of tests may be run at different speeds, and always with a wide open throttle. This will result in maximum torque at each speed and increasing horsepower up to some limiting speed. Beyond this limiting speed, the horsepower decreases. Or, a similar series of tests may be run at a partial throttle opening. Or, the torque and speed may be kept constant (corresponding to average load conditions) and the air-fuel ratio varied, or changes in the spark timing made, or temperature of inlet or outlet water, inlet air, etc. Of course, when any such series of tests is decided upon, it is very necessary that all other quantities be kept as

constant as possible. In all cases curves of the results should be plotted.

The brake horsepower of automotive engines is often estimated by empirical formulas, especially for the purpose of rating. The formula, given below, was originated by the Association of Licensed Automobile Manufacturers and was later adopted by the Society of Automotive Engineers. At the present time it is used only for purposes of rating engines. The formula is

$$\text{B.hp.} = \frac{D^2 \times N}{2.5},$$

D being the bore in inches, and N the number of cylinders. This rating applies to four-cycle engines at an assumed piston speed of 1000 ft. per min.

(b) **Brake Mean Effective Pressure.** Because of the difficulty of indicating the modern high-speed automotive engine, it is practically impossible to determine the real mean effective pressure. Since the B.hp. = Mech. Eff. × PLAN ÷ 33,000; the product, **Mechanical Efficiency × Mean Effective Pressure** = B.hp. × 33,000 ÷ PLAN. The product is measurable, as shown by the terms it is equal to, and is called the "Brake Mean Effective Pressure."

(c) **Fuel Consumption, Gallons or Pounds per Brake Horsepower-Hour.** The general principles of this determination are the same as for other fuel consumption tests except as regards the *measurement* of the fuel. The S.A.E. Test Standards require that all fuel consumptions shall be measured by noting the decrease in weight of a tank from which fuel is being fed to the carburetor. The tank must be placed on a sensitive platform and connected to the carburetor with a length of tubing. The tubing should be sufficiently flexible in order that it will not interfere with the weighing. The weighings are made in the following manner:

The counterpoise of the scale-beam is set so that the scale-beam will fall just as the test is started. The counterpoise is then set back to the next half-pound or pound mark. The time required for the weight of fuel to be consumed is noted and from these data the weight per hour can readily be computed.

(d) **Thermal Efficiency and Heat Balance.** These subjects are treated in the same manner as in Test 57.

(e) **Friction Horsepower.** Where an electric dynamometer is available, it is possible to obtain the approximate friction horsepower of the engine. The dynamometer is used to drive the engine at various speeds and the torque reaction is measured. The reaction will, of course, be in

the opposite direction to that obtaining when the engine is driving the dynamometer. Most dynamometers are provided with a linkage system which changes the direction of the pull so that the scales will always give the reading no matter which way the torque reacts.

The friction horsepower test should be made while the engine is still warm, preferably immediately after the brake horsepower test. During the test the throttle should be in the *same position as during the brake horsepower test*. If the brake horsepower has been made at various throttle openings, a series of friction horsepower tests must be made at these same throttle openings.

(f) Correction of Brake Horsepower to Standard Conditions. Tests of automotive type engines are frequently run under widely varying conditions of atmospheric pressure and temperature. In order to place such tests on a uniform basis the Society of Automotive Engineers has set up *standard conditions* which are 29.92 in. of mercury (standard pressure) and 520° F. absolute (standard temperature). The correction formula is

$$\text{B.hp.}_c = \text{B.hp.}_o \times \frac{P_s}{P_o} \times \sqrt{\frac{T_o}{T_s}}$$

where B.hp.$_c$ = corrected brake horsepower;

B.hp.$_o$ = observed brake horsepower;

P_o = observed barometric pressure in inches of mercury;

P_s = standard barometric pressure of 29.92 in. of mercury;

T_o = observed absolute room temperature in degrees F.;

T_s = standard absolute temperature of 520° F.

SECTION II

STEAM BOILERS

60. Economy Test of a Steam Boiler

Principles. The heat available in the coal consumed in the furnace of a steam boiler is distributed under four heads: First, the useful heat which goes to evaporate the feed water; second, the heat lost with the carbon slipping through the grate or removed in cleaning; third, the heat carried up the stack by the exhaust gases; and fourth, the heat lost by radiation and all otherwise unaccounted for losses. The heat carried up the stack may be traced under the subheads, heat represented by temperature of the dry exhaust gases, enthalpy of steam formed by evaporation of water in the coal and from the combustion of hydrogen, and heat lost through incomplete combustion of carbon, hydrogen, and hydrocarbons.

A complete boiler test to determine these quantities includes measurements of feed water and coal supplied for a counted time, and analyses of the coal, ash, and flue gases.

Besides the heat balance, certain other quantities are to be found for the purpose of comparing the general performance of the boiler with that of others. The following terms should be understood in this connection.

A *unit of evaporation* (U.E.) is an output of 1000 B.t.u. per hr. in the steam. This unit has been adopted by the A.S.M.E. Test Code for Stationary Steam Boilers and bears no direct relation to the equivalent evaporation. It is simply a convenient unit of output.

Equivalent Evaporation. If steam were generated in a boiler at atmospheric pressure from feed water at 212° F. it would be necessary to add only the latent heat, 970.4 B.t.u., to produce 1 lb. of dry and saturated steam. When steam is generated at any other pressure from feed water at any other temperature, the number of B.t.u. added to each pound actually evaporated divided by 970.4 will give the number of pounds which would have been evaporated from feed water at 212° F. and at atmospheric pressure. This is called the equivalent evaporation from and at

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212° or just equivalent evaporation. It is very useful in comparing the performance of boilers operating at widely differing conditions of pressure and feed-water temperature.

Boiler Horsepower. When the equivalent of 34.5 lb. of steam per hr., from and at 212°, is generated, the boiler is said to have an output of one boiler horsepower (1 Bo.hp.). In the modern boiler test codes, this unit has been discarded since it is meaningless. Boilers are now rated on their output *in B.t.u. per hour* or *units of evaporation* which is thousands of B.t.u. per hour. This method of rating is not, as yet, universally used and many boilers will be found still rated according to the old plan. Old ratings may be converted to the new method of rating as follows:

$$U.E. = \frac{Bo.hp. \times 34.5 \times 970.4}{1000}.$$

Boilers are also rated by the number of pounds of steam evaporated per hour under stated conditions of pressure, superheat and feed-water temperature.

The rate of combustion for coal-fired furnaces is the number of pounds of coal consumed per hour per square foot of grate surface. This expression, however, is not applicable to furnaces burning powdered coal, liquid or gaseous fuels. The rate of combustion in these cases is expressed as the *number of B.t.u. liberated per cubic foot of furnace volume per hour*. This method of expressing combustion rate is sometimes useful in connection with stoker-fired furnaces.

Duration of test is governed by the error of starting and stopping. All the conditions of the boiler and furnace should be the same at the end of the test as they were at the beginning, especially in regard to the quantity and quality of coal on the grate, and water in the boiler. If the coal remaining on the grate at the end of the test is greater than that on the grate at the start, the furnace will be charged with a greater amount than it has actually consumed, and vice versa, unless an estimate is made of the difference between the amounts. The probable error of such an estimate is greater in the case of the coal than of the water. Consequently, the error in the coal measurement determines the duration of the test. The test should be long enough to make the error of starting and stopping less than 1 or 2 per cent of the total amount of coal fired. Assuming this error to correspond to a difference of 1 in. in the fuel bed, the resulting error in the coal quantity will be 2.5 lb. for each square foot of grate surface, the weight of a cubic foot of incandescent coal being about 30 lb. Two and one-half pounds is 1 per cent of 250 lb.,

so the test should be continued until a total of 250 lb. of coal per square foot of grate surface has been fired. The higher the rate of combustion, the shorter is the duration. If the rate is 20 lb. per sq. ft. of grate per hour, for example, the test should be about 12 hr. long.

In all cases, the length of the test should be a multiple of the regular cleaning period in order to cover and keep constant regular operating conditions. In the example just cited, if the cleaning period is 8 hr., the test should be continued for 16 hr.

Starting and Stopping. The boiler should be operated during a preliminary run and complete readings taken until they show uniformity of all conditions. The furnace should then be thoroughly cleaned, enough live coal being left on the grate, as in usual cleaning, to start a new fire. The thickness of this coal bed should be estimated and noted, and the level of the water in the water column marked with a string tied around the glass. The boiler is then fired with a fresh charge of weighed coal, the time of firing the first shovelful of which is taken as the start of the main test. The ash-pit should then be cleaned and all temperatures, pressures, etc., read. At the end of the test, the cleaning is repeated in exactly the same way as at the beginning, the time of firing the first shovelful of the fresh coal after cleaning being taken as the end of the test. The same observations are made as at the beginning. It is best to note the water level just after the cleaning, with the fire door open, because then there will be a minimum of ebullition to cause a false level.

Sampling of the coal should be done as for the proximate analysis. Sampling of the flue gas is described under Products of Combustion, Part II. Sampling of the ashes and refuse removed in cleaning and falling through the grate is according to the same principles as for coal. The steam should be sampled for quality determinations.

To make clear the quantities resulting from a boiler test, a set of calculations from observations made for an actual test will be given for each quantity analyzed in the following. The condensed or average observations are given below.

DATA FROM A BOILER TEST AND NOTATION USED IN FORMULAS

Duration of test, hours	$= 12$
Total weight of coal as fired, pounds	$= 4,925$
Total weight ash and refuse, pounds	$= 734$
Total weight water evaporated, pounds	$= 36,400$
Boiler pressure, pounds per square inch, abs.	$= 110$
Quality of steam, x	$= 0.992$
t_f = temperature of feed water, degrees F.	$= 98$

DATA FROM A BOILER TEST AND NOTATION USED IN FORMULAS
(Continued)

t = temperature boiler room, degrees F. .... = 80
T_e = temperature flue gas, degrees F. .... = 680
Proximate analysis.　Percentages by weight of coal as fired.
Moisture .... = 2.75
Volatile matter .... = 6.00
Fixed carbon .... = 78.45
Ash .... = 12.8
Weight of carbon in 1 lb. of refuse .... = 0.20
m, vm, fc = weights of moisture, volatile matter, and fixed car-
　　　　　　bon per pound of dry coal, respectively.
Heat value of coal by calorimeter .... = 13,040

VOLUME ANALYSIS OF THE FLUE GAS, PER CENTS

D = carbon dioxide .... = 11.0
O = oxygen .... = 9.0
M = carbon monoxide .... = 0.5
N = nitrogen .... = 79.5

WEIGHTS, IN POUNDS, OF CARBON FROM 1 LB. OF DRY COAL

C_a = carbon wasted in ash;
C_t = total carbon; fixed, and in volatile matter;
C_g = carbon burned, appearing in flue gas.

(a) Determination of Total Coal and Refuse.　The coal to be fired is weighed out by the barrowful as it is needed and heaped on the boiler room floor at a place convenient for firing, one barrowful at a time.　The following is the best form of log for coal records.　It includes a few observations to illustrate its purpose.

Weight of Barrow		Net Weight	Total Weight	Time of Firing First Shovelful
Empty, Lb.	Full, Lb.			
Test started	...	...	...	8:00 A.M.
100	250	150	150	8:00
100	275	175	325	8:24
100	260	160	485	8:46
Etc.				

As the test proceeds a chart should be plotted of total coal as shown by the fourth column against time as a base. It should be noted particularly that the time to be plotted against the total weight from each barrowful is not the time of firing the first shovelful from the corresponding weighing of coal, because this coal is not entirely burned until the first shovelful of the next charge is fired. Referring to the log, 150 lb. should therefore be plotted against 8:24; 325 lb. against 8:46; etc. If the rate of evaporation and all other conditions external to the furnace are uniform, then with proper firing, the rate of combustion will be uniform, and a fair straight line will be represented by the points of the plot. This line should pass through the origin; if it intersects the axis of coal weights above the time axis, it indicates that there was too much coal on the grates at the start; if below, too little. In such a case, it is better to figure the total coal from the slant of the line.

FIG. 110.—Weighting System.

The ashes accumulating under the grates and the refuse removed during cleanings in the main test should be collected, allowed to cool without wetting, and then weighed.

In the modern boiler plant, of any considerable size, the boilers are fired by means of stokers or with pulverized coal. In practically all cases, equipment is provided for continuous weighing of the coal as it is fired and the weighing is carried out automatically. This simplifies the matter of weighing but, in stoker-fired furnaces, it is not always easy to be certain that the condition of the fuel, in the furnace, is the same at the beginning as at the end of the test. In furnaces fired by pulverized coal this problem is not present since the coal burns almost as soon as it enters the furnace and there is no large mass of fuel present at any time. This is also true of furnaces which are fired with oil as a fuel and similarly with gas.

(b) **Determination of Total Feed Water Evaporated.** The feed water may be measured by a meter in the feed-water line, but when certainty of accurate results is desired, a direct weighing system such as represented diagrammatically by Fig. 110 should be used. With this arrangement, the level of the water at the start of the test is marked by the gage, g, in the suction tank and by the string on the water column at m. The feed

pump should be operated at a constant rate through the test, this rate being determined by the horsepower required of the boiler; and, with proper firing and regulation, the water level shown by the column should always be approximately at m. The best form of log for the records is as follows. A few sample observations are included to illustrate.

Barrel Number	Weight of Barrel		Net Weight	Total Weight	Time of Passing Gage Level
	Empty, Lb.	Full, Lb.			
Test started	..	...	...	...	8:00 A.M.
1	50	300	250	250	8:12
2	60	320	260	510	8:25
1	75	300	225	735	8:34
Etc.					

As the test proceeds, a chart should be plotted of total water as shown by the fifth column against the time taken to evaporate it as a base. When the first charge, 250 lb. on the log, is turned into the suction tank, the level is raised above the gage, g, Fig. 110. When the level has again descended to the gage, all of the 250 lb. has then been evaporated provided that none of it has accumulated in the boiler as would be shown by a higher level in the column than m. The time of passing the gage level in the suction tank is therefore the time to be plotted; 250 lb. against 8:12; 510 lb. against 8:25; etc.

It is especially important to keep the rate of evaporation uniform for a valid test. The water-time curve should therefore be a straight line.

If the water level is not the same at the end as at the beginning of the test, the total water may be figured from the slant of the line, or a correction may be applied to the last figure in the fifth column of the log. The correction is calculated from the cubical contents of the boiler between the two levels and the density of the water at the existing temperature. It should be noted that a false level may appear in the water column if the ebullition is violent or if the water column is blown down within a short time before reading. The latter occurrence is due to the fact that blowing down forces into the glass hotter and less dense water than is there normally.

(c) **Quantities to Be Found Prior to Calculations of Results.** The proximate analysis of the coal is generally expressed in percentage by weight of the coal "as fired," that is, including moisture. To find the weights per pound of dry coal, it is necessary only to divide each of the items by 100 minus the percentage of moisture. Thus, for the data of the test previously cited,

$$m = 2.75 \div (100 - 2.75) = 0.0283 \text{ lb. of moisture;}$$
$$vm = 0.0617 \text{ lb. of volatile matter;}$$
$$fc = 0.807 \text{ lb. of fixed carbon;}$$
$$a = 0.132 \text{ lb. of ash;}$$
$$vm + fc = 0.869 \text{ lb. of combustible.}$$

It is necessary to know the **total carbon per pound of dry coal.** If the ultimate analysis has been made, this figure is available; otherwise, it is necessary to estimate it. The method of making this estimate is outlined in the section on **proximate analysis.** Using the data of the above analysis, the total weight of carbon in 1 lb. of dry coal, by this method, is

$$C_t = 0.807 + 0.01 \times 0.869 = 0.816 \text{ lb.}$$

The proportion of **carbon** which is **wasted through cleaning and grate,** C_a, may be found in two ways. The first method is as follows:

The total carbon in the refuse equals the total amount of refuse minus the ash from the total coal fired. The ash from the total coal fired equals the total quantity of coal times the proportion of ash in it as given by the proximate analysis. Then,

$$C_a = \frac{\text{Total refuse, lb.} - \text{total coal, lb.} \times \text{per cent ash} \div 100}{\text{Total coal, lb.} \times (1 - \text{per cent moisture in coal} \div 100)},$$

the denominator being the total weight of dry coal.

The second method uses the proximate analysis of the ash.

$$C_a = \frac{\text{Total refuse, lb.} \times \text{weight of C in 1 lb. refuse}}{\text{Total coal, lb.} \times (1 - \text{per cent moisture in coal} \div 100)}.$$

In the first equation it is assumed that all of the ash coming from the coal fired during the main test is removed with the refuse. This, however, is uncertain, so it is better to use the second method. Applying the data of the test under consideration, each pound of the refuse was found to contain 0.20 lb. of carbon. The total coal as fired was 4925 lb. and

it contained 2.75 per cent of moisture. The total weight of refuse was 734 lb. Then

$$C_a = \frac{734 \times 0.20}{4925 - 0.0275 \times 4925} = 0.0307 \text{ lb.}$$

To find the **weight of the carbon burned.**

$$C_g = C_t - C_a.$$

For given data

$$C_g = 0.816 - 0.0307 = 0.785.$$

(d) To calculate the total equivalent evaporation the heat added to a pound of feed water is first found. Referring to the test data given and the steam tables, the heat added is

$$305.5 + 0.992 \times 882.5 - (98 - 32) = 1115.$$

The total equivalent evaporation is the result of dividing the product of the total water, and the heat added per pound by 970.4,

$$\text{E.e.} = \frac{1115 \times 36,400}{970.4} = 41,860 \text{ lb.}$$

The equivalent evaporation per pound of coal, as fired, is found by dividing the total equivalent evaporation by the total weight of coal fired.

$$\text{E.e.c.} = \frac{41,860}{4925} = 8.48 \text{ lb.}$$

Per pound of dry coal

$$\text{E.e.d.c.} = \text{E.e.c.} - (1 - \text{moisture, lb.})$$

$$\text{E.e.d.c.} = \frac{8.49}{1 - 0.0275} = 8.73 \text{ lb.}$$

Equivalent evaporation per hour

$$\text{E.e. hr.} = \frac{41,860}{12} = 3489 \text{ lb. per hr.}$$

Units of evaporation

$$\text{U.E.} = \frac{3489 \times 970.4}{1000} = 3385.$$

All the items of the heat balance will be figured on the basis of 1 lb. of dry coal. The heat value of the dry coal as found by the calorimeter or calculation will therefore be the heat supplied.

(e) **Over-All Efficiency.** Since there are E.e.d.c. $\times$ 970.4 B.t.u. added to the feed water for each pound of dry coal fired, and since the heat available from this coal is its heat value, the efficiency is,

$$\frac{\text{E.e.d.c.} \times 970.4}{\text{Heat value of dry coal}}.$$

$$\frac{8.73 \times 970.4}{13,040} = 0.65 \text{ or } 65\%.$$

(f) **Loss to Carbon Wasted in Refuse.** Taking the heat value of carbon as 14,600 B.t.u., this loss per pound of dry coal is

$$14,600 C_a.$$

For the given data, $14,600 \times 0.0307 = 448$ B.t.u., or $448 \div 13,040 = 3.4\%$.

(g) **Loss of Heat to Dry Exhaust Gases.** This determination involves the measurement of the excess of temperature of the flue gas over that of the boiler room, and of the weight of dry flue gas per pound of dry coal, W_d. For the specific heat, an average value, 0.24, may be used with sufficient accuracy. Then the loss is

$$0.24 \times W_d \times (T_e - t).$$

To find W_d the formula $W_d = W_a + C_g$ may be used. For the given data

$$W_d = \frac{3.04 \times 79.5 \times 0.785}{11.0 + 0.5} + 0.785 = 17.3 \text{ lb. per lb. of dry coal}$$

and the loss is $0.24 \times 17.3 \times (680 - 80) = 2490$ B.t.u. or $2490 \div 13,040 = 19.1\%$.

(h) **Loss of Heat to Water Vapor in the Flue Gases.** To calculate this, we must know the weight of vapor per pound of dry coal and its heat content above room temperature, including latent heat. The vapor carried in as humidity of the air need not be considered because it is small in quantity and its latent heat is not added by the coal. This leaves only the vapor from the moisture in the coal and that due to combustion of hydrogen to consider. From the formulas for W_v, the weight of vapor per pound of coal, and for the heat of steam when mixed with flue gases, we have

Loss per pound dry coal $= (9H_t + m)(1090 + 0.46T_e - t)$.

H_t, the weight of hydrogen per pound of dry coal, may be determined according to Test 43(b) in the case of semi-anthracite and bituminous coals if the ultimate analysis has not been made. For the proximate analysis of the given test, the calculation for H_t is made, resulting in $H_t = 0.025$. Therefore the heat lost is $(9 \times 0.025 + 0.0283)$ $(1090 + 0.46 \times 680 - 80) = 335$ B.t.u. or $335 \div 13,040 = 2.6\%$.

(i) **Loss of Heat Due to Incomplete Combustion.** Considering only the incomplete combustion of carbon, the loss of heat when 1 lb. of carbon burns to CO instead of CO_2 is $14,600 - 4400 = 10,200$ B.t.u. In the boiler test there are W_i lb. of carbon per pound of dry coal burned to CO. The loss is

$$10,200 \times W_i \text{ B.t.u. per pound of dry coal.}$$

For the given data,

$$\text{Heat lost} = 10,200 \times \frac{0.5 \times 0.785}{11 + 0.5} = 338 \text{ B.t.u. or } 338 \div 13,040 = 2.7\%.$$

(j) **Radiation** is found by difference. For the given data, it is

$$100\% - 65\% - 3.4\% - 19.1\% - 2.6\% - 2.7\% = 7.2\%.$$

(k) **Other Quantities.** The boiler horsepower is, as previously defined,

$$\frac{3489}{34.5} = 101 \text{ B.hp.}$$

The equivalent evaporation per hour per square foot of heating surface is,

$$\frac{\text{E.e.hr.}}{\text{Sq. ft. of surface}}.$$

For the given boiler, the heating surface was 1000 sq. ft., so E.e.hr. per square foot $= 3489 \div 1000 = 3.49$ lb. or U.E.

The grate surface of this boiler was 30 sq. ft., so the **rate of combustion** of dry coal, as previously defined, was $399 \div 30 = 13.3$ lb. per hr. per sq. ft.

Efficiency of the Grate and Cleaning. The proportion of the combustible lost is $C_a \div (vm + fc)$. Consequently this efficiency is,

$$1 - \frac{C_a}{vm + fc}.$$

For the given data, this is $1 - 0.0307 \div 0.869 = 96.5\%$.

Efficiency of the boiler and furnace (exclusive of grate) is the over-all efficiency divided by the efficiency of the grate. For the given data, this is $65 \div 96.5 = 67.4\%$.

This efficiency is often defined as the heat absorbed per pound of the combustible burned divided by the heat valve per pound of combustible.

The excess coefficient may be found from the expression derived in section on exhaust gas analysis.

Besides the total water and coal curves, it is well to plot, after the test, all the readings of pressure, feed-water, flue and boiler-room temperatures, per cent CO_2 and the draft in inches of water. These will be broken curves. They will aid deductions of causes of any irregularities in the performance.

A few barometer readings are desirable in connection with natural draft.

All circumstances that may affect the conditions of the test should be carefully noted for possible future reference.

61. Heat Balance Test of Gas or Oil-Fired House Heater

Principles. This class of tests is becoming of increasing importance because of the extensive application of the apparatus to heating of dwelling houses and other buildings.

The heating is generally accomplished through the circulation of low pressure steam, hot water, or, so-called "vapor," that is, steam at a pressure less than atmospheric. Since it is very difficult to measure the quality of low-pressure steam from these heaters, it is preferable to make laboratory tests with the apparatus used as a hot-water heater even though it be designed to generate steam. The useful heat transferred is a little greater under this condition than when steam is made, first because the water heating surface is greater when there is no steam space, and, second, because the temperatures of the H_2O are less which assists heat transfer and lessens radiation loss.

Although the operation in service is intermittent because of thermostat control, laboratory tests are made under continuous operation, the rate of fuel flow being the most important independent variable.

Adjustment should be made, at the burner, for the best air-fuel ratio, as indicated by flue gas analysis, before each test is started.

Duration of test may be as short as 1 to 2 hr., since, if water only is heated, there is no uncertainty as to weight contained at start and finish, and there is no error of the same nature in fuel measurement.

Starting and Stopping. The heater should be run sufficiently long after preliminary adjustments to establish uniform conditions. Test data may be taken at any time thereafter.

When gas is the fuel, a conventional dry meter may be used with less than 1 per cent of error. With oil burners, the fuel may be syphoned from a 2- to 5-gallon open can placed on a scale of sufficient precision to give significant readings at 5- to 10-min. intervals.

(a) **Heat supplied** with fuel gas should be figured exactly as for a gas engine, Test 57(a) and (b). The volume of gas, by meter, is reduced to standard cubic feet, and its heating value determined on the same basis. If the heat balance is calculated on the basis of 1 standard cu. ft. of dry gas, the heat supplied is its heating value.

For oil burners, on a like basis, the heat supplied from the fuel is the heating value in B.t.u. per pound of oil. These burners have as auxiliaries a pilot flame operating on gas, and an electric motor to supply air and spray the fuel. To the heating value of the oil should be added the heating value of that part of a cubic foot of gas supplied per pound of oil. With many arrangements all of the electrical energy supplied to the motor enters the furnace in the form of heat. This also should be accounted for in the heat supplied, as well as in costs of operation.

(b) **Heat utilized** is that transferred to the water. The water may be obtained from the city mains, and weighed upon leaving the heater. Feed-water pumps are not a part of regular equipment since house heaters depend upon gravity for water supply.

Thermometers accurate to within 1° should be used, and placed as nearly as possible, in inlet and outlet, to the boiler shell.

(c) **Sensible and latent heat lost to exhaust gases,** with gas fuel, may be figured exactly as for a gas engine, Test 57(f), (g), and (h). For oil fuel, the calculation is as for coal-fired furnaces, except that the "gasified carbon," C_g, may be taken as equal to the "total carbon," C_t.

When the stack temperature is less than 600°, the specific heat of the dry gases may be taken as 0.24. Gas-fired heaters of good design give very low stack temperatures, 300 to 350°. Oil-fired heaters sometimes equal this, but more frequently their stack gas temperatures run higher.

(d) **Loss to Incomplete Combustion.** There is generally complete combustion with gas fuel in properly adjusted and designed burners. A test for CO should not be omitted, nevertheless. With oil fuel, if there is not good atomization and turbulence in the combustion chamber, the loss to unburned fuel may be very large. It is well to check this by making a rough heat balance during test. If the loss to radiation and "unaccounted

for" is greater than 1 to 5 per cent (as low as 1 per cent for radiation is quite normal), then a dry sample of flue gas should be collected for complete analysis. By "dry sample" is meant one that is not allowed to come in contact with water. The analysis should include determinations of carbon monoxide, hydrogen, methane, and illuminants (C_2H_4). Small percentages of these combustibles, in the exhaust, mean a large heat loss.

(e) Rating and Other Determinations. House heaters are generally rated in terms of the number of square feet of house radiator surface they can supply. If steam is used, it is assumed that 240 B.t.u. per hr. can be transferred, for heating purposes, from each square foot of radiator surface. Consequently,

Rating in sq. ft. of "steam radiation" = Heat utilized in B.t.u. per hr. $\div$ 240.

The excess coefficient in the combustion calculations is found as in gas engine testing, for gas fuel, and as under boiler testing for oil fuel.

SECTION III

TESTING OF AUXILIARY EQUIPMENT

62. Test of Centrifugal Pumps

Principles. The useful work of a centrifugal pump is the product of the weight of water delivered in a given time and the total head against which the water is pumped. The weight of water is measured in pounds and the head is measured in feet. The total head is the sum of the pressure head, the velocity head, and the head due to height with reference to some chosen datum; usually the datum is taken as the center line of the pump.

The work supplied to the pump is that received by its shaft in the case of a belt-driven machine, or one directly connected to an electric motor. When the pump is driven by direct connection to a steam engine or turbine, both pump and drive are generally tested as a single unit as would be a steam-driven reciprocating pump (see Test 65).

The principles employed in connection with the test of a centrifugal pump also apply to propeller pumps.

Selection of the Independent Variable. The rotative speed, quantity discharged, or head may be varied and used as independent variables. Since most centrifugal pumps are driven by constant-speed motors, it is usual to select the quantity discharged as the independent variable and to take head, horsepower and efficiency as dependent variables in separate sets of curves. In any case, when one variable is changed arbitrarily, another is kept constant and the third becomes the dependent variable.

Variation may be obtained in the quantity discharged by placing a throttle valve in the discharge line. No throttle valve should ever be placed in the suction line.

When the head is kept constant, the speed is varied by control of the driving motor. Each change of speed must be accompanied by a change in the valve opening in the discharge line in order to maintain the constant head.

(a) Measurement of Head. The total head produced by the pump is represented by the difference between the absolute heads at the discharge

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and suction nozzles, taking into account any difference in the elevations of these nozzles with respect to a fixed datum, and any difference between the velocity heads at the points of measurement. The fixed datum usually taken is the center line of the pump.

The standard method of measuring head is by means of a water column or gage glass which gives a direct reading. When it is not possible to use this method, an indirect method may be employed such as a mercury manometer or a Bourdon gage. The end of the tube, connecting the gage with the water conduit or pipe, should be flush with the inside of the conduit in which the pressure is to be measured. The gages described show the static pressure only; it does not indicate the velocity head. The velocity head may be computed since the size of the conduit and the quantity are known. The total head then can be expressed as follows:

When a mercury manometer is used, a correction must be made for the column of water in the leg of the manometer connected to the pipe. When Bourdon gages are used, their indications are for the center of the gage; if the gage is installed above the pipe, a correction must be made for the column of water in the pipe connecting the gage to the water line. This is taken as the vertical distance from the center of the water line to the center of the gage. A similar condition exists when the gage is below the pipe in which the pressure is to be measured but the correction is of the opposite sign, positive in this case.

The total head, against which a pump is working, is most conveniently expressed as follows:

$$H_t = \left(H_d + \frac{V_d{}^2}{2g} + z_d \right) - \left(H_s + \frac{V_s{}^2}{2g} + z_s \right)$$

where H_t = total head, feet;
$\quad H_d$ = discharge head, feet;
$\quad H_s$ = suction head, feet;
$\quad V_d$ = velocity in discharge line, feet per second;
$\quad V_s$ = velocity in suction line, feet per second;
$\quad z_d$ = distance from center line of discharge line to center line of pump, feet.
$\quad z_s$ = distance from center line of suction line to center line of pump.

In the last two items the distance is always measured vertically and the sign of the quantity becomes negative if z is measured downward from the center line of the pump.

A more general expression for the total head is expressed in the more usual terms of the Bernoulli equation:

$$H_t = \left(\frac{P_d}{w} + \frac{V_d^2}{2g} + z_d\right) - \left(\frac{P_s}{w} + \frac{V_s^2}{2g} + z_s\right)$$

where P_d = pressure in the discharge, pounds per square foot;

P_s = pressure in suction line, pounds per square foot;

w = specific weight of liquid pumped, pounds per cubic foot.

Care must be exercised in all of these expressions to keep the algebraic signs correct. For example, if P_s is below atmospheric pressure, it takes a negative sign.

Where water is being pumped it is convenient to recall that pressure, in pounds per square inch, may be converted to head in feet by multiplying the pressure by $2.31 = 144 \div 62.4$. This does not apply to liquids which are more or less dense than water.

(b) **Capacity.** The capacity of a centrifugal pump may be expressed in gallons or cubic feet per minute, or in gallons or cubic feet per 24 hr. This latter expression is usually given as millions of gallons per day (m.g.d.). The methods of measuring the rate of discharge are the same as those described under tests of hydraulic turbines.

(c) **Water horsepower** may be calculated from the relation

$$\text{w.hp.} = \frac{W \times H}{33,000},$$

where W = weight of water discharged per minute;

H = total head.

(d) **Horsepower Supplied.** The most accurate method of measuring the power supplied is by driving the pump with an electric dynamometer. If no dynamometer is available, the pump may be driven by a calibrated electric motor. (See section on tests of electric machinery.) If the pump is driven by a steam engine or turbine, it is more convenient to find the horsepower supplied to the set rather than the pump alone.

(e) **Efficiency.** The efficiency of the pump may be calculated by dividing each value of water horsepower by the corresponding value of power supplied.

(f) **Corrections to Constant Speed.** When it is not possible to maintain a strictly constant speed, values obtained at *slightly* different speeds may be corrected to the standard speed selected. These corrections are based on the fact that, within limits, the capacity varies directly as the speed, the head varies as the square of the speed and the shaft horsepower varies as the cube of the speed.

(g) Curves may be plotted from the data obtained, as follows: For constant speed tests, power supplied, capacity, and efficiency are plotted against head. For constant head tests, the same items are plotted against speed. Sometimes the speed or head, power supplied, and efficiency are plotted against capacity.

In tests of considerable magnitude, covering complete runs at a number of different speeds, it is informative to superimpose the curves of constant efficiency upon the family of head-capacity curves. These iso-efficiency curves take on the aspect of a series of contour lines and show completely the performance of the pump over the entire working range of speed.

63. Test of a Rotary Pump

Rotary pumps of the displacement type, such as a gear pump, are tested similarly to centrifugal pumps. The volumetric displacement may be computed for each speed. The difference between this computed volumetric capacity and the actual capacity found by test may be stated as slip in per cent of the displacement.

64. Valve Setting of a Duplex (Steam) Pump

Principles. The duplex pump consists of two reciprocating steam pumps operating side by side and arranged so that the piston rod motion of one operates the steam valve of the other. Each pump consists of a water cylinder whose piston is directly driven by the piston rod of the steam cylinder.

The steam valves distribute the steam so that one pump is on its forward stroke when the other is on its return. The valves have neither

Fig. 111.—Duplex Pump Valve Motion.

laps nor lead, so that cutoff occurs simultaneously with release. The valve stem of each pump is driven by a rocker deriving its motion from the piston rod of the other pump (see Fig. 111). As each rocker moves through the whole piston rod stroke and as it is desirable that the valves throw only at the end of the stroke, a certain amount of lost motion is provided between the valve stem and the valve. This is shown by the distance c in the figure. The valve stem passes freely through a hole in a lug on the valve, and gives motion to the valve only when the nuts n

and n come into contact with the lug. By adjusting the lost motion, the valve may have more or less throw.

There are two steam and two exhaust ports for each steam cylinder, SS and EE. This is arranged as a safeguard against the piston hitting the cylinder head. If the valve should not close the exhaust port before the end of the exhaust stroke, the piston itself will close it, thus entraining steam in the clearance space which acts as a cushion.

(a) **Adjustment of Lost Motion of Valve.** This is accomplished by setting the valves in their mid-positions when the pistons are in their mid-positions.

The pistons should be set by reference to prick marks on the piston rods. The limiting positions of stroke should be measured by prying each piston rod with a bar to the extreme of its travel, pressure behind the pistons being released by opening drains or otherwise. When the piston rods are set midway between these positions, the rockers should be vertical.

With the steam chest cover removed, the valves are now placed so that they completely cover the ports. The nuts n, n, Fig. 111, are then adjusted so that there is an equal amount of clearance between each one and the valve lug. With some designs, there is only one nut to be adjusted on each stem, this nut then being between two lugs. In such a case the clearances may not be changed in amount, but only equalized. In other designs there is a "lost motion link" outside of the valve chest by which the clearances may be adjusted without opening the chest.

The proper amount of clearance depends upon the size and design of the pump and may be determined by trial. If the steam ports are to be opened full each stroke, the total lost motion should be equal to the throw of the valve stem corresponding to normal piston stroke less twice the steam port length parallel to the stroke.

(b) **Readjustment after Trial.** If, after the valves are set, the pump runs "lame," that is, the strokes are unequal in length or time, it may be because the resistances of the water cylinders are different. They should then be examined for leaky water valves, tight stuffing boxes, etc. Such faults being corrected, if the pump still runs lame, the lost motion of the valves must be readjusted until the desired equality of strokes is obtained. It need only be remembered that to diminish the lost motion makes the valve throw greater, thus providing better access to the steam and release to the exhaust.

65. Mechanical Efficiency Test of a Reciprocating Steam Pump

Principles. The mechanical efficiency of a steam pump equals the useful or water horsepower, w.hp., divided by the indicated horsepower of the steam cylinders. If H is the total head in feet pumped against, including suction and discharge as in Test 62, and W is the number of pounds of water delivered per minute, then,

$$\text{w.hp.} = \frac{WH}{33,000}.$$

If there were no leakage of water in the pump from one side of the piston to the other or through the valves, then the work done per minute would be equal to the product of the total pressure pumped against in pounds per square foot and the volume displaced by the piston in cubic feet per minute. If p stands for the pressure in the discharge pipe; p_1, for the vacuum in the suction pipe, both in pounds per square inch; and if L and A are the same as in the formula for indicated horsepower, and N is the number of working strokes per minute, then

$$\text{w.hp.} = \frac{144(p + p')LAN}{33,000 \times 144}.$$

On account of leakage this formula is not a reliable one with which to determine the water horsepower. It may be used only by assuming a value for the leakage, and this may be very inexact.

Leakage of water in a pump, of the kind referred to, is called "slip." It is generally expressed numerically as a percentage of the piston displacement.

(a) **Steam End Indicated Horsepower at Various Water Horsepowers.** The indicated horsepower is found exactly as for a steam engine (see Test 50(a)), except in regard to measurement of the stroke length, L. With direct acting reciprocating pumps, there is no constant limit to the stroke length and in ordinary operation it may vary materially. An average value should therefore be obtained from a number of measurements throughout the test. There is on the market a continuous recording device for this purpose, but if one is not available, the average stroke may be obtained from the average lengths of the indicator diagrams, the ratio of reduction being known. Another method is to set a scale against a mark on some projecting part of the piston rod, and to note the travel of this mark at regular time intervals.

The water horsepower may be varied either by changing the speed of the pump or the discharge pressure. With reciprocating steam pumps, the pressure is usually approximately constant, and the water load varies with the quantity of water demanded. Either the pressure or the speed may be the independent variable during test, according to the operating conditions of the pump.

The pressure may be varied during the test for different runs, by opening the stop valve in the discharge pipe more or less.

The quantity of water delivered per minute may be measured by any of the methods of determining water rates. For large pumps, weirs may be used.

The head against which the water is discharged may be measured with a pressure gage set in the discharge pipe. The suction head generally may be measured in feet, and taken as the difference in level between that of the water supplied and the center of the pressure gage used for measuring discharge pressure. Since 1 lb. per sq. in. = 2.31 ft. head, the total head is

$$H = 2.31p + H_s$$

in which H_s is the suction head in feet.

Readings of p and H_s should be taken at regular intervals during each run.

These quantities determine the water horsepower according to the formula previously given. The water horsepower may be estimated, according to the second formula, if the discharge pressure and the number of strokes per minute are measured, and if a value for the percentage of slip is assumed (see (d)).

Neither of the formulas for water horsepower takes into account the kinetic energy of the water delivered. Generally the velocities of the water are low enough to make this a negligible quantity. If it is desired to calculate the horsepower due to kinetic energy, the velocity should first be figured from the cubic feet discharge per unit of time and the cross-section of the pipe. The result is then obtained by,

$$\text{w.hp. available from kinetic energy} = \frac{Wv^2}{2g} \div 33,000$$

in which W has the same value as before, v is the velocity in feet per second, and $g = 32.2$, nearly.

(b) Mechanical and Fluid Losses. The loss due to mechanical friction of the pump parts may be obtained by indicating both the water and

steam ends of the pump. The indicated horsepower of the water end is obtained in exactly the same way as for the steam cylinder, the same values being used for the number of strokes per minute and the average length of stroke. Then the steam end I.hp. minus the water end I.hp. equals the mechanical friction horsepower.

The fluid losses are due to leakage, or slip, fluid friction of the water against its passage in the ports, eddies, etc. Slip causes loss of power through water being pumped against pressure from one part of the pump to another without being discharged into the delivery main. These losses are included in the power shown by the indicator diagram for the water end. Consequently the horsepower lost equals the I.hp. of the water end minus the w.hp. as calculated under (a).

(c) **The gross efficiency** is obtained by dividing the w.hp. by the corresponding I.hp. of the steam end. It is sufficient, for each run, to use average values of heads, water rates, mean effective pressures, and stroke lengths and speeds with which to calculate a single value of efficiency.

The indicated horsepower of the water cylinders is sometimes measured. This divided by the I.hp. of the steam cylinders is the efficiency covering mechanical losses only.

(d) **Slip.** The piston displacement in cubic feet is $\dfrac{LAN}{144}$ per minute. The water actually displaced, in the same units, is $W/62.4$ at ordinary temperatures. The percentage of slip is therefore,

$$\frac{\dfrac{LAN}{144} - \dfrac{W}{62.4}}{\dfrac{LAN}{144}} \times 100 = \left(1 - \frac{2.3W}{LAN}\right) \times 100.$$

The quantities involved have been obtained for the other determinations.

(e) **The capacity** of pumps is generally expressed in gallons per 24 hr. Knowing the weight of water, or cubic feet, discharged per minute, the capacity is readily calculated, for which purpose the following closely approximate relations may be used.

1 gal. weighs 8.33 lb.   1 cu. ft. contains 7.48 gal.

Rate of discharge of a pump may be measured in the manner described for centrifugal pumps. Since the rate of most steam pumps is fairly low, it is usually best to weigh the discharge water in calibrated or weighing tanks.

66. Economy Test of a Steam Pump

Principles. These, in general, are the same as given for the economy test of a steam engine, Test 51, the only difference being that the useful work is measured in terms of water horsepower as defined under Test 65.

In addition to the measurements listed under Test 51, another one is usually required, namely, of "duty." Duty is the number of foot-pounds of useful work performed by a pump per million B.t.u. consumed by the engine.

(a) Steam and Heat Consumption. The pounds of steam consumed per hour may be found by any of the methods under Test 51 (a). The water horsepower is determined as under Test 65 (a). From these two quantities the pounds of steam per w.hp.-hr. may be calculated.

The heat consumed per pound of steam may be found by using the same notation and reasoning given under Test 51, Principles. Pressure readings of the supply and exhaust steam, and quality determinations of the supply steam are required for these heat quantities.

(b) Thermal Efficiency. This is the same as for Test 51 (c) except that the Rankine standard need not be used in most cases.

(c) Duty. The thermal efficiency (as stated above), when expressed as a fraction, is the heat equivalent of the useful work done per B.t.u. consumed. The number of foot-pounds of useful work done by each B.t.u. consumed is therefore 778 times the efficiency, 778 being the mechanical equivalent of heat. Since the duty is the number of foot-pounds per million B.t.u. consumed,

$$\text{Duty} = 778,000,000 \times \text{Efficiency}.$$

It should be noted that the efficiency is here expressed not as a per cent, but as a fraction of the heat consumed.

67. Test of a "Power" Pump

Principles. By "power" pump is meant a reciprocating pump driven by a crank shaft which receives its power through a belted pulley, or gears from a motor. The general principles to be studied for testing are similar to these previously outlined under Test 65.

(a) Water Horsepower. See Tests 65 (a) and 62 (c).

(b) Available Horsepower. This may be found as for a centrifugal pump, Test 62 (b), when the drive is by belt or geared motor.

(c) **Mechanical and Fluid Losses and Efficiencies.** The difference between the water horsepower and the indicated horsepower equals the hydraulic losses. The difference between the indicated horsepower and item (b) equals the mechanical losses. Expressions for the corresponding efficiencies are obvious. The total efficiency is item (a) divided by item (b).

(d) **Slip and Capacity.** See Test 65 (d) and (e).

68. Economy Test of an Injector

Principles. The injector is a pump in which the heat energy of steam is directly used. Works on steam engines and steam boilers describe the instrument in detail. The results from an injector test are the same as those from a steam pump test. In addition, the number of pounds of water pumped per pound of steam supplied is usually quoted. The methods of testing are somewhat different, inasmuch as a direct measurement of the weight of steam per hour is not necessary, this quantity being obtained from the heat balance.

Fig. 112 shows diagrammatically the arrangement of an injector for test.

An expression for the weight of steam used per minute is deduced by equating the heat energy of the steam entering the injector, plus the mechanical and heat energy of the water entering, to the heat and mechanical energy of the water discharged. The datum of the heat measurements is at 32° F., and of the energy measurements is the level of the injector.

Fig. 112.—Injector Test.

Let W = weights, in pounds per minute;
   H_f = heat of the liquid;
   H_{fg} = heat of vaporization of the steam supplied;
   x = quality of the steam supplied, if wet;
   p = pressure of discharge, pounds per square inch;
D', D'' = suction head, and distance between injector and gage, feet, respectively;
   t = temperatures, degrees F.

Subscripts f, s, d refer to feed water, steam, and discharge, respectively. Then, expressing all energies in heat units, and neglecting kinetic energy
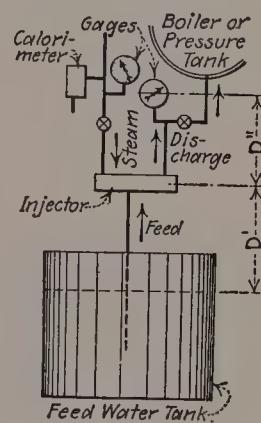
$$W_f h_f - \frac{W_f D'}{778} + W_s(H_f + H_{fg}) = W_d h_d + \frac{W_d(2.31p + D'')}{778} + \text{radiation.}$$

Also,

$$W_d = W_f + W_s.$$

Solving these equations simultaneously and neglecting radiation, we have

$$W_s = \frac{H_{fd} - H_{fs} + \dfrac{2.31p + D''}{778} + \dfrac{D'}{778}}{H_{fs} + xH_{fg} - H_{fd} - \dfrac{2.31p + D''}{778}} \times W_f.$$

In this expression, the heat equivalents of the work are very small compared with the other quantities and they may be omitted. For the heat of the liquid, h, in each case, may be substituted its closely approximate value, $t - 32$. There results,

$$W_s = \frac{t_d - t_f}{t_s + xH_{fg} - t} \times W_f.$$

Selection of the Independent Variable. This may be the discharge pressure, the feed-water temperature, the suction head, or the steam pressure. When the injector is used as an ordinary pump, the discharge pressure is perhaps the most important of these. If this is selected as the independent variable, it may be controlled by means of the valve in the discharge pipe, see Fig. 112. If the steam pressure is selected, it may be varied by the stop valve in the steam pipe, but the gage should be placed on the other side of the valve to record the pressure on the injector. Whichever quantity is selected, all of the others should be kept as constant as possible throughout each run, and the injector should be regulated before each run so as to secure maximum flow.

(a) **Pounds of Water Pumped per Pound of Steam.** It must first be decided whether to credit the injector with the feed water only, or with the feed water plus the condensed steam which appears in the discharge. If the injector is used for *emptying*, as when draining a mine, the former amount is logically used; if for *filling*, as when supplying a tank at a height, the latter amount is proper. When used as a boiler feed, there is room for some discussion of which is the correct amount. The condensed steam is returned to the boiler, but it is used again to operate the injector, so strictly speaking it is not a part of the useful feed water. On the other hand, a reciprocating steam pump used for feed water would *not*

return the steam used to drive it and would have to pump that much more water.

From the equation previously deduced, we have

$$\frac{W_f}{W_s} = \frac{t_s + xH_{fg} - t_d}{t_d - t_f},$$

which is the pounds of feed water pumped by 1 lb. of steam. Adding one to the numerical value of this expression gives the pounds of water plus steam pumped.

The temperatures are obtained from thermometers appropriately placed in the discharge, feed, and steam pipe; the latent heat, from the steam tables; and the quality, from a calorimeter. The injector should be run long enough before each run at the required conditions to establish uniform temperatures. The run should be long enough to secure enough temperature readings for a fair average, and for the proper measurement of the quantities mentioned later.

(b) **Pounds of Steam per Water Horsepower-Hour.** The equation for W_s, previously deduced, gives the pounds of steam used per minute. The water horsepower is

$$\text{w.hp.} = \frac{W_f(2.31p + D' + D'')}{33,000} \quad \text{or} \quad \frac{W_f D' + W_d(2.31p + D'')}{33,000},$$

the one or the other to be used according to the considerations given under (a). The pounds of steam per water horsepower-hour are then

$$S = \frac{60W_s}{\text{w.hp.}}.$$

For this quantity it is necessary to make the same measurements as enumerated under (a) and also to measure the feed-water rate and the heads, D' and D''. The pounds of feed water per minute may be determined by any of the methods for measuring water. A convenient arrangement is to use a hook gage or water glass in the tank from which the feed is obtained, by which the drop in water level may be timed. This causes a variation in the suction head, but if the tank is of generous cross-section, the percentage of variation may be small enough not to affect the results materially. The average value of the suction head may be figured from the readings of water level. The head D'' is, of course, constant throughout the test.

(c) Thermal Efficiency. When considered merely as a pump, the same expression for efficiency as for a steam engine holds, Test 51 (c).

$$\text{Efficiency} = \frac{2545}{S(H_{f_s} + xH_{fg} - H_{fa})}.$$

When considered as a boiler-feed apparatus, the efficiency is much higher than this because the heat in the discharge is then "useful," and the only loss is that due to radiation. A determination of radiation may be made from the heat balance if the steam supplied is measured as well as the feed water, from which the boiler-feed efficiency may be found. This requires much more precision in all the test measurements than the others results do, however, because the radiation is a small difference between two large quantities.

(d) Duty. See Test 66 (c).

(e) Capacity should be figured in gallons per 24 hr. from the test results of W_f or W_d, according to the use to which the injector is to be put.

69. Economy Test of the Pulsometer

Principles. The pulsometer is a pump in which the pressure energy of steam is applied directly to the water pumped without intermediate pistons. It consists of two cast-iron chambers controlled by automatic valves and working alternately. Steam, entering one, discharges the water by its superior pressure, while in the other, the partial vacuum, formed from steam condensed by contact with the water, draws a fresh supply of feed.

The results from a pulsometer test, the equations to be used, and the method of testing are exactly the same as for an injector test, Test 68, to which the reader is referred.

70. Test of a Surface Condenser

Principles. A steam condenser receives, in practice, a mixture of steam and water and air. The steam entering the condenser, if exhausted from a steam engine or turbine, carries with it moisture. Air enters the system through leaks in steam-pipe joints, stuffing boxes, and from the boiler feed.

Since it is the function of a condenser (considered as a power unit) to maintain a vacuum, this is one of the most important items to investigate. Under ideal conditions of heat transfer, the vacuum possible to be

maintained would be that of saturated steam corresponding to the temperature of the outgoing condenser water. This ideal is not realized for two reasons. First, a *difference* of temperature between the steam and the cooling water is required that there should be a heat flow from the one to the other. Second, the effect of air mixed with the steam is to make a higher pressure than that due to the steam alone, according to the law of partial pressures. If, for example, the temperature in the steam space of a condenser were 126° F., the corresponding steam pressure (from the tables) is 2 lb. absolute, very nearly. Suppose there is air present in amount equal to one-quarter, by weight, of the steam. A cubic foot of the steam space would then contain a weight of air equal to one-quarter of the density of steam at 2 lb., absolute, and the specific volume of the air would be four times that of the steam, that is, $4 \times 174 = 696$ cu. ft. per lb. Consequently (from $PV = RT$).

$$\text{Pressure of the air} = \frac{53.4 \times (126 + 460)}{144 \times 696} = 0.31 \text{ lb. per sq. in.}$$

and the pressure of steam and air combined would be $2 + 0.31 = 2.31$ as compared with 2 lb. if the steam space contained steam alone. It follows, then, that the effect of air upon condenser performance increases the absolute pressure above that corresponding to the prevailing temperature, and therefore lessens the effectiveness of the condenser.

The question of the most economical vacuum is a pertinent one in connection with condenser tests. This leads to a consideration of the cost of cooling water, and of auxiliary power, that is, of circulating, wet-air, and dry-air pumps.

Finally, the rate of heat transmission should be investigated in order to determine the effectiveness of the cooling surface.

The independent variable may be the vacuum, weight of condensate, or the quantity of cooling water per unit of time.

The following notation will be used in this and the next test:

W_w = weight of cooling water, pounds per hour;
W_s = weight of wet steam entering, pounds per hour;
W_r = weight of cooling water per pound of condensate;
T_s = temperature of the steam in the condenser, degrees F.;
t_s = temperature of the condensate discharged;
T_i = temperature of cooling water at inlet, degrees F.;
T_o = temperature of cooling water at outlet, degrees F.;

T_m = mean temperature difference between steam space and cooling water, degrees F.;

B = heat transmitted, B.t.u. per hour;

A = area cooling surface, square feet.

(a) **Ideal and Actual Vacuums.** The actual vacuum may be measured with a calibrated gage, its indications being reduced to absolute pressure by subtracting them from the barometric pressure as observed. Or, better yet, an "absolute pressure gage" may be used. This may be in the form of a glass U-tube with one leg sealed at the top and completely filled with mercury. A sufficient reduction of pressure on the open leg will lower the mercury column in the filled one; the difference of level in the two legs of the U-tube then is the absolute pressure. A heavy rubber tube, capable of withstanding 15 lb. collapsing pressure, may be used to connect the open leg of the U-tube to the condenser steam space. The steam inlet pipe should be tapped, close to the condenser, for a $\frac{1}{8}$-in. pipe nipple. Another opening should be made for the insertion of a thermometer.

If there is but little air in the steam, and the pressure drop through the condenser is small, the absolute pressure as shown by the gage should be but slightly greater than that of saturated steam corresponding to the average temperature of the steam space or of the discharged condensate. A large quantity of air leakage, on the other hand, will be indicated by a material difference between these pressure values.

(b) **Conditions of Operation.** The condenser should be operated, as in regular service, at a predetermined vacuum. The quantity of cooling water and the speed of the auxiliaries should also be predetermined and, together with the vacuum, should be maintained constant. Temperature readings should be taken of the steam entering the condenser and of the discharge from the wet air pump or hot well pump.

A series of tests may be run at various values of the vacuum. It is useful to plot the resulting data, actual against ideal vacuum (corresponding to both T_s and t_s), for the purpose of later comparisons.

(c) **Starting, stopping and duration** of condenser tests should follow the same general rules as those for steam engine and turbine tests. For frequency of readings, the general rules should apply.

(d) **Quantity of Steam Condensed.** This can be measured best by collecting and weighing the condensate and making such corrections for condenser leakage as are necessary (see Test 71).

(e) **Rate of Heat Transmission.** This may be found by estimating the heat content of the entering steam, or by finding the heat added to the

cooling water. The water quantities are usually large, and therefore some form of velocity meter is appropriate for measuring the water supplied. Inlet and outlet temperatures of the cooling water (close to the condenser) must be read and averaged. From these data may be calculated the heat transferred per hour.

$$B = W_w(T_o - T_i).$$

(f) **Effectiveness of Heat Transmission.** This can be judged after determining the number of B.t.u. transmitted per square foot of surface per hour per degree difference in temperature. This result represents the conductivity of the heat path, and will be referred to as C. Then

$$C = \frac{B}{A \times T_m}.$$

The mean temperature difference, T_m, is found from the relation

$$T_m = \frac{T_o - T_i}{\log_e \dfrac{T_s - T_i}{T_s - T_o}},$$

which is deduced in treatises on heat transmission.

A second method of calculating the temperature difference is the arithmetic mean:

$$T_m = T_s - \frac{T_o - T_i}{2}.$$

The logarithmic mean temperature difference is based on the assumptions that (1) the steam temperature is constant throughout the steam space and (2) the heat transfer is uniform in all parts of the condenser. The arithmetic mean temperature difference is based on the assumptions that (1) the steam temperature is uniform throughout the steam space and (2) there is a uniform rise of the temperature of the cooling water from inlet to outlet. Fairly complete tests have shown that the temperature is not uniform throughout the steam space and the rate of heat transfer is not uniform. Where the rise of temperature of the cooling water is small, it is probable that the arithmetic mean temperature difference is sufficiently accurate and some authorities advise its use as being more representative of actual performance. Where large temperature rise of the cooling water is encountered, it is best to adhere to the logarithmic mean temperature method since it then more nearly approximates actual conditions.

Substituting the expression for T (log) and the one previously quoted for B in the equation for C, we have

$$C = \frac{W_w}{A} \times \log_e \frac{T_s - T_i}{T_s - T_o}.$$

The value of C resulting from this equation should be about 300 B.t.u. for ordinary installations, and, in the best designs, as high as 900 B.t.u. Low values indicate fouled surfaces, air pocketing, flooding, etc.

It may be noted from an examination of the equation for C that the more nearly equal are T_s and T_o, the more effective is the heat transmission. These temperatures, by themselves, are therefore a criterion of performance.

Another statement of the effectiveness of heat transmission is in terms of weight of condensate per square foot per hour. This is not very definite, since it depends upon the enthalpy of vaporization of the steam and consequently the vacuum, as well as the quality of the steam. Ordinarily it is about 10 lb.

(g) **Weight of Condensing Water per Pound of Condensate.** Under ideal conditions, with dry steam, this would be

$$w_r' = \frac{L}{T_s - T_i}.$$

Actually it is

$$w_r = \frac{xH_{fg} + (T_s - t_s)}{T_o - T_i}$$

plus an amount necessary for cooling the entrained air, radiation, etc. The value of w_r may be calculated from this equation, provided an estimate of x is made, and compared with the ratio of weights of cooling water to condensate as obtained by metering both.

71. CONDENSER LEAKAGE TEST

Principles. The pressure in the steam space of a condenser being below atmosphere, cooling water will find its way into the steam space through any leaks which may exist. Any such leakage increases the apparent amount of condensate and thus constitutes an error in the measurement of the steam rates of the steam engine or turbine to which the condenser is attached. In order to eliminate this error, it is necessary to determine the rate of leakage under the conditions of operation. There are five approved methods:

(a) **Qualitative silver nitrate method** is *only* suitable for cooling water containing dissolved salts, such as sea water or tidal mixtures of fresh and salt water. The method is simply an indication of whether or not the condenser is leaking and depends on the reaction of chlorides with silver nitrate; the precipitation of a cloudy white silver chloride. The method can sometimes be used with *fresh water* if the water can be artificially chlorinated before entering the condenser.

(b) **Quantitative silver nitrate method** is used *only* with salt or brackish cooling water. When method (a) indicates that there is leakage, this method is used to determine the amount or rate. Briefly, it consists in titrating a sample of the condensate and cooling water against a standard solution of silver nitrate using pure condensed steam as a control. The leakage rate is then found by computation.

(c) **Electrolytic-conductance method** compares the conductivity of the condensate and cooling water, again using pure condensed steam as a control. The method requires special apparatus and considerable skill in making the determination. The method is not usable where the cooling water contains less than *10 grains* of dissolved solids per gallon. Leakage rates are found by computation.

(d) **Short direct weight method** is generally sufficient for ordinary laboratory tests. It is to be used *only with fresh cooling water* and with the engine or turbine, connected to the condenser, *not running*. All steam supply must be completely shut off. If there is any doubt as to the tightness of valves, the main steam line should be disconnected. When it is certain that no steam can leak into the steam space, to give a false reading, the test may proceed.

The circulating pump and air pumps are run at the same speed as under normal operating conditions and the same vacuum is maintained as nearly as possible. All water removed by the wet vacuum pump is circulating water which has leaked into the steam space. This is collected for a period of time and weighed. From these data, the rate of leakage may be calculated.

(e) **Standard direct weight method** is the same as the *short direct weight method* except for the addition of refinements which increase the accuracy. It is to be used *only* with fresh water in the circulating system.

The A.S.M.E. Test Code on Leakage Measurement should be consulted for full details on all of the above tests.

72. Test of a Jet Condenser

Principles. These are, in the main, the same as for the preceding Test 70. There is this difference, however, that, since condensate and cooling water are mixed, the outlet temperature of the latter, T_o, equals that of the former, t_s.

(a) **Ideal and Actual Vacuums.** See Test 70(a).

(b) **The quantity of steam condensed** may be found in several ways:

1. By taking the steam, supplied to the connected unit, from an isolated boiler or group of boilers, the quantity of steam condensed can be found by measuring the feed water and correcting this measurement for drips, boiler and piping leakage, and for steam not supplied to the unit.
2. Steam supplied to the unit can be measured by a steam meter, if such practice is considered sufficiently accurate. The meter must be calibrated in place under actual service conditions.
3. If the condition of the steam exhausted to the condenser can be determined, its amount can be determined from the heat given to the cooling water, **provided the amount of cooling water is known.**

(c) **Rate of Heat Transmission.** The method of Test 70(b) may be used with the modified formula,

$$B = W_w(t_s - T_i).$$

If the weight of the discharge from the condenser is measured instead of the inlet water, W_w, then steam entering the condenser must be separately determined and subtracted from the total weight to find W_w.

(d) **Weight of Condensing Water per Pound Condensate.** See Test 70(g). In the formulas quoted t_s may be substituted for T_o since the two are the same.

73. Test of a Feed-Water Heater

Principles. As far as heat transmission is concerned, a closed feed-water heater is identical to a surface condenser; and an open heater to a jet condenser. Tests 70 and 72 should therefore be read in this connection.

The closed feed-water heater generally operates with the water space under pressure, but the open heater runs practically at atmosphere. The steam supplied may be engine or auxiliary exhaust. There may be more steam available than can be used for preheating the water, in which case

the excess is either vented to the atmosphere or used in some other apparatus, such as radiators for room heating.

There are so many different combinations for the employment of feed-water heaters that it is impracticable to state a general method of testing covering them all.

The same notation will be used in this test as was used for condensers. It should be understood, in this connection, that the words "feed water" should be substituted for "cooling water," and "feed-water heater" for "condenser."

(a) **Useful Heat Transmitted.** For a closed heater this is, in B.t.u. per hour,

$$B = W_w(T_o - T_i).$$

For an open heater, and for a closed one in which the condensate is returned to the boiler, there should be added to this the heat regained in the condensate.

The quantities in the equation just quoted are to be measured as under Test 70(e).

(b) **Effectiveness of Heat Transmission.** For a closed heater Test 70 (f) applies. In either type, the more closely the outlet temperature of the water approaches that of the steam, the more effective is the heat transmission.

(c) **Available Enthalpy.** It is useful, sometimes, to estimate the enthalpy available in the total steam supplied the heater (above feed-water temperature) in order to form an idea of the proportion of it that is utilized.

74. Test of a Reciprocating Air Compressor

Principles. In the operation of an air compressor, it is the purpose to increase the pressure of the air supplied so as to make available the energy it contains. As this energy is to be used when the air is cool, it is desirable to compress the air isothermally. If it is allowed to heat, the pressures during compression are higher and more work is required for compression. For this reason, water jackets and intercoolers are used, the heat removed by them being a saving.

Referring to Fig. 113, the dotted lines show an ideal air compressor diagram from a cylinder without clearance in which the air is discharged at a pressure, 4_2–5_1, equal to that in the delivery main. The supply is drawn in at atmospheric pressure along the line 6_1–2_1 and then compressed isothermally along 2_1–4_2.

The actual indicator diagram varies from the ideal one as shown by the full lines, 1–2–4–5. The friction resisting the motion of the air through inlet and outlet valves and ports necessitates a lesser pressure than atmospheric to draw in the air, and a greater pressure than that in the delivery main to discharge it. Consequently, work is lost as represented by the areas 4–5₁–5 and 1–2–3–6. The actual compression line, 2–4, must be above isothermal because of the impossibility of perfect cooling, and this results in the loss represented by the area between the lines 2–4 and 2–4₁. These are "compression" losses.
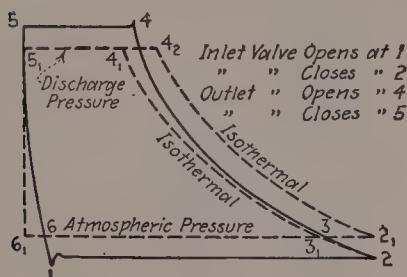
FIG. 113.—Ideal and Actual Air Compressor Diagrams.

When the outlet valve closes at point 5, an amount of air equal to the clearance volume is entrained in the cylinder. At the beginning of a new cycle, this air expands so that the effective stroke for drawing in air begins at point 6 instead of 6₁. This point occurs later, the higher the delivery pressure is at point 5. When the inlet valve closes at 2, the air in the cylinder is rarefied because of the suction, and it is not until 3 is reached that its pressure becomes atmospheric. The effective stroke for drawing in air is therefore represented by line 6–3 instead of the full length of the diagram. This is a "volumetric" loss, since less air is delivered than would be if the full stroke were effective. Another volumetric loss, not appearing in the diagram, is due to leaky valves and pistons, similar to slip of a pump.

After the air in the delivery pipe has been cooled down to room temperature, it contains energy, due to its pressure above atmospheric, available for performing work. If this air could be expanded isothermally to its original pressure, there could be regained all of the energy necessary to compress it isothermally. This, however, cannot be done, the actual expansion being more nearly adiabatic. There is a final loss, then, due to the lack of availability of all the energy added to the air.

These various losses are indicated in certain expressions for efficiency, to be defined in the following, which are generally figured as air compressor test results.

In defining the various items, the following notation will be used:

Q = cubic feet of air discharged per minute, as measured;

T = absolute temperature of air discharged, degrees F.;

P = absolute pressure of air discharged, pounds per square inch;
t = absolute temperature of air supplied, degrees F.;
p = absolute pressure of air supplied, pounds per square inch;
γ = ratio of specific heats;
N = number of stages in a multi-stage compressor.

Indicated air horsepower is obtained in the same manner as the indicated horsepower of a steam engine, being figured from the mean effective pressures of the indicator diagrams of the air cylinders.

Net air horsepower is the horsepower required to compress, isothermally, the measured quantity of air from room pressure and temperature to the pressure of the delivery main. It may be computed from the following equation:

$$\text{Net A.hp.} = \frac{144Q}{33,000} \, p \log_e \frac{P}{p}.$$

Air Horsepower to Compress Adiabatically. This may be computed by means of the following equation. This equation serves both for single and multi-stage compressors by substituting the proper value for N in the expression,

$$\text{A.A.hp.} = \frac{144\gamma NQ}{33,000(\gamma - 1)} \, p \left[\left(\frac{P}{p} \right)^{\frac{\gamma-1}{N\gamma}} - 1 \right].$$

The gross horsepower is the power supplied to the compressor. In the case of a steam or internal-combustion engine driven compressor, it is the indicated horsepower in the steam or power cylinders. In the case of an electric motor drive, it is the output horsepower of the motor; the electrical horsepower multiplied by the motor efficiency.

The mechanical efficiency equals the indicated air horsepower divided by the horsepower supplied to the compressor.

The volumetric efficiency is the cubic feet of free air actually delivered in a given time divided by the low pressure piston displacement in the same time.

The compression efficiency is found by dividing the horsepower required to compress adiabatically all of the air or gas delivered by the compressor by the indicated air horsepower.

The over-all efficiency is the ratio of the net air horsepower to the gross horsepower.

(a) **Capacity** may be expressed as the number of cubic feet of air discharged per minute at the pressure in the delivery main corrected to room temperature, or as the number of cubic feet discharged per minute re-

duced to free air, that is, air of the same temperature and pressure as that supplied to the compressor.

The capacity of a compressor in terms of compressed air is

$$Q_1 = Q \times \frac{t}{T},$$

and in terms of free air is

$$Q_2 = Q \times \frac{t}{T} \times \frac{P}{p} = Q_1 \times \frac{P}{p}.$$

T and t are to be determined by thermometers, one in the room near the air inlet to the compressor; the other in the discharge pipe just beyond the last cylinder or receiver. p is obtained by a barometer and reduced to pounds per square inch; P, by a pressure gage set close to the thermometer measuring T.

The expression of the capacity in terms of free air is the one most commonly used.

The methods of measuring air and gas flow described in Part I are used to determine the amount of air or gas compressed. The formulas for these instruments result in the number of pounds of air or gas flowing per unit of time. The results obtained from such computations and measurements must be converted to terms of volume usually at the conditions of pressure and temperature existing at the point of intake. This conversion is most conveniently accomplished by use of the characteristic gas equation

$$PV = WRT$$

in which P = absolute pressure at intake, pounds per square foot;
$\quad\quad\quad T$ = absolute temperature at intake;
$\quad\quad\quad R$ = gas constant, 53.3 for air;
$\quad\quad\quad W$ = weight of gas flowing, pounds;
$\quad\quad\quad V$ = volume at intake conditions, cubic feet.

With some gases, such as natural gas, the value of R is not sufficiently constant for reliable results. In such cases a value for R should be chosen which best fits the range of pressures encountered.

(b) **Mechanical Efficiency.** The gross air horsepower is figured from the indicator diagrams from the air cylinders. The horsepower supplied to the air compressor, if it is steam driven, is the indicated horsepower of the steam cylinders. If it is belted or geared to the source of power, the horsepower supplied is that received at the compressor shaft, and should

be obtained as for a fan blower, Test 75 (a). From these results the mechanical efficiency may be obtained.

(c) **Volumetric Efficiency.** The length of the line 6–3, Fig. 113, divided by the length of the diagram 6_1–2_1, is often referred to as the "apparent" volumetric efficiency; apparent, because it does not take account of slip and leakage. The "true" volumetric efficiency is,

$$Q_2 \div L(A_h + A_p)N$$

in which L is the length of the stroke in feet, A_h and A_p are the areas in square feet of the piston at the head and power ends, respectively, and N is the number of double strokes per minute. If the compressor is single-acting, A_p should be omitted.

If the compressor has more than one cylinder, the expression applies to the low-pressure cylinder.

(d) **Compression Efficiency.** It is first necessary to compute the horse-power required to compress the given volume of air adiabatically. The methods of making this computation has been outlined under definitions. Dividing this value by the indicated air-horsepower gives the efficiency of compression.

(e) **Overall Efficiency.** (See definition.)

(f) **Separation of Losses.** A precise separation of all the losses cannot readily be made for the reason that they merge into each other. For example, mechanical friction in the air cylinder adds heat to the air which makes the loss due to ineffective cooling greater. If there is leakage of air past the inlet valve during the compression, the compression curve will approach more nearly to the isothermal, making an apparent gain in the jacket's efficiency, but an actual loss as far as power and capacity are concerned. Again, if the inlet valve is mechanically operated and opens too early, the compressed air in the clearance space escapes and the apparent volumetric efficiency is improved, but power is lost.

(g) **Moisture Corrections.** When the air or gas compressed contains moisture, some of which is removed during the processes of intercooling and aftercooling, the weight of the air and water vapor mixture passing the metering device will be less than that entering the compressor. In converting the measure of the air compressed, as found by the metering device, to terms of intake pressure and temperature, a correction must be applied for the condensed water vapor removed.

There are two methods for making this correction; *correction through vapor pressure* and *correction through vapor density*. Both of these

methods are too complicated to be stated here but may be found, in full, in the A.S.M.E. Test Code for Displacement Compressors and Blowers. Both methods should be employed in order to check results. *In any case the correction is quite small.*

75. Test of a Fan Blower

Principles. There are two kinds of fan blowers, called "pressure" and "volume"; the one delivering air at high pressure, the other at high velocity. The shape of the fan casing and rotor to a large extent determines its kind.

The capacity of a blower depends upon the volume of "free air" that it will discharge in a given time at a given rotative speed. By free air is meant air at the pressure and temperature of the room at the time of the test. Capacity is generally expressed in cubic feet per minute.

The useful work done by a fan equals the energy imparted to the air as pressure and velocity. If W is the number of pounds of air discharged per minute, and H_1 is the pressure expressed as a head of air in feet, then WH_1 is the foot-pounds of useful work per minute represented by the pressure energy. If V is the velocity in feet per second, then $WV^2 \div 64.4$ is the foot-pounds of useful work represented by the kinetic energy. The quantity $V^2 \div 64.4$ is the head of air, in feet, equivalent to the velocity, or "velocity head," which will be referred to as H. Then the useful or "air-horsepower" is

$$\text{A.hp.} = \frac{WH_1 + WH}{33,000}.$$

Selection of the Independent Variable. In the operation of a fan blower, either the rotative speed, the velocity of air, or the pressure of the air may be varied. The first is controlled according to the type of the driving engine. The air velocity or pressure may be varied at a given rotative speed only by changing the size of the outlet from the air discharge pipe. A useful set of tests may be had by making the rotative speed the independent variable. The pressure is kept constant during a series of runs at different speeds; then the pressure is changed for another series, and so on. In order to keep the pressure constant when the speed is changed during each series, it is necessary to adjust the external resistance to the air. This may be done by using different nozzles or orifices as outlets from the discharge pipe. A convenient

arrangement consists of a leaf shutter, similar to that of a camera, which may be readily changed to any desired diameter of outlet orifice.

A set of curves may be made between the various results and the rotative speeds. For each pressure there will be a corresponding set of curves. Taking the results from these curves on a coordinate representing one value of the speed, another set of curves may be plotted with pressure as the independent variable. A number of such sets may be made corresponding to various speeds.

Another independent variable which is sometimes used is the size of the orifice or nozzle in the outlet pipe. Pressure and velocity will then vary at each different speed, and these with the other variables may be plotted against orifice area; one set of curves for each speed of the blower.

The duration of each run need not be more than a few minutes if a velocity meter is used for air measurement. The observed quantities vary very slightly, as a rule, during each run, so that only two or three sets of observations are needed.

(a) Determination of Horsepower Supplied. If the fan is belt driven, a transmission dynamometer may be used to advantage. Allowance should be made for belt losses, since the horsepower supplied to the fan shaft is desired. This may be done by using the revolutions per minute of the fan shaft with the torque shown by the dynamometer to calculate horsepower, and multiplying the result by the ratio of diameters of fan pulley to dynamometer pulley. This allows for belt slip.

If the fan is driven by an electric motor, the horsepower supplied may be had from a calibration of the motor, readings of the current supplied then being necessary.

(b) Capacity. For the measurement of the air quantity a meter of the velocity type is most readily adapted. Of these, anemometers, orifices, and Pitot tubes are customarily used, preference being given to Pitot tubes.

Velocities may be obtained from the Pitot tube either by making a complete traverse of the pipe at each determination, or by locating the velocity opening at the point of mean velocity. Multiplying the velocities by the cross-sectional area of the pipe gives volumes of air under the pipe conditions of pressure and temperature. To reduce these results to free air, pressures and temperatures of the air in the pipe and room must be read. In the pipe, the pressure may be read from a manometer attached to a branch from the tube leading to the static opening of the Pitot meter. For room pressure, the barometer should be read.

Consider as an example the following readings. Velocity head, $h = 0.5$ in. of water; pressure = 4 ins. of water; barometer = 29.9 ins. of mercury; temperature of room = 65° F.; temperature of air in pipe = 67° F. The absolute pressure of the air in the pipe is $29.9 + 4 \div 13.6 = 30.2$ ins. of mercury or $30.2 \times 0.49 = 14.8$ psi.

The absolute temperatures of the room and of the air in the pipe are 525° and 527°, respectively. The density of the air in the pipe may be figured from the familiar relation,

$$w = \frac{144p}{53.4T} = 2.7 \frac{p}{T},$$

which gives $w = 2.7 \times 14.8 \div 527 = 0.0758$ lb. per cu. ft. To convert the velocity head h, in inches of water, into H, in feet of air, we have,

$$H = \frac{62.3}{0.0758} \times \frac{h}{12} = 68.5h,$$

for the given data. Consequently, $H = 68.5 \times 0.5 = 34.25$, and the velocity is

$$V = 8.02\sqrt{H}$$

$$V = 8.02\sqrt{34.25} = 47 \text{ ft. per sec.}$$

If the area of the pipe is 0.33 sq. ft., the cubic feet of air per minute is $60 \times 0.33 \times 47 = 930$. To reduce this to free air, multiply by the ratio of temperatures and the inverse ratio of pressures:

$$\text{Free air per minute} = 930 \times \frac{525}{527} \times \frac{30.2}{29.9} = 934 \text{ cu. ft.}$$

The Standard Test Code for Centrifugal Fans and Blowers, adopted by the National Association of Fan Manufacturers and the American Society of Heating and Ventilating Engineers, requires the reduction of all test results to *Standard Air* rather than to *free* air at room temperature and pressure. Standard air is air weighing 0.07488 lb. per cu. ft. This weight corresponds to air at a barometric pressure of 29.92 in. of mercury, a dry-bulb temperature of 68° F., and a relative humidity of 50 per cent. *Observance of this requirement of the code places all fan and blower tests on the same basis.*

(c) **Horsepower Supplied per Thousand Cubic Feet of Free Air per Minute.** This quantity is obtained from the results of (a) and (b) being the quotient between each two corresponding values multiplied by 1000.

(d) Air Horsepower. For the data previously given, the weight of air delivered per minute is $930 \times 0.0758 = 70.5$ lb. The velocity head is 34.25 ft. of air, and the pressure head, $68.5 \times 4 = 274$ ft. of air. Then the air horsepower is

$$\text{A.hp.} = \frac{70.5(34.25 + 274)}{33,000} = 0.659.$$

(e) The Efficiency equals the air horsepower divided by the horsepower supplied.

SECTION IV

COMPLETE PLANT TESTING

76. Economy Test of a Steam Power Plant

Principles. A complete test consists of measurements of the amounts of heat distributed and lost in the entire system, beginning with the energy of the fuel and ending with the energy delivered at the shaft or switchboard. For the present we shall consider only that part of the plant beyond the boiler, the subject of boiler trials being separately treated under Test 60.

The most important result from power plant tests is the fuel consumption in pounds of fuel per horsepower- or kilowatt-hour. Other results are the steam and heat consumption of the engine and the auxiliaries, and the heat balance. The latter is an equation between the total heat available and the various amounts accounted for in its distribution. They may be found in heat units per hour and expressed as per cents of the total heat per hour added to the feed water entering the boiler. If these percentages are separately multiplied by the boiler efficiency, the items of the heat balance will then be percentages of the heat available in the fuel. This follows from the fact that boiler efficiency is that part of the fuel energy which is added to the feed water and distributed as steam.

The boiler trial, which is part of the complete test and conducted at the same time, gives the efficiency of the boiler and its losses, all based on the heat value of the fuel. A complete heat balance is thereby obtainable.

The usual condensing power plant contains as auxiliaries a condenser, circulating pump, air pump, and feed-water pump. There may also be a cooling tower and its auxiliaries, and a feed-water heater using the exhaust steam from the auxiliaries, and such minor apparatus as separators, traps, etc., which may or may not be arranged to drain back to the boiler.

The rearrangement of the apparatus and piping for the purpose of testing should interfere as little as possible with normal operating con-

ditions, and any departures from them that may affect the results, such as different feed-water temperatures, should be allowed for by making a separate test to measure the actual working quantities that have been altered.

Equipments of different power plants are too varied for very specific directions for the measurements of the various quantities. Any of the methods given under Test 51 (a) for the measurement of steam weights should be applied according to the exigencies of the particular test. Any one item of the heat balance may be omitted from the direct measurements, since one item may always be figured by subtraction.

The basis of all heat measurements is the heat contained by the feed water just before it enters the boiler or economizer if there is one.

Duration. If a boiler trial is included, its duration determines the length of the test; if the engine and auxiliaries only are to be tested, the duration should be the same as for a test of the engine alone. (See Tests 51 and 60, principles.)

(a) Steam and Heat Consumption of Engine and Auxiliaries per Horsepower- or Kilowatt-Hour. Let w stand for weight of steam per hour; H, for its total enthalpy per pound above 32°, depending upon its pressure and quality; and let h be the enthalpy per pound of the feed water above 32°. Let the subscripts e, c, a, f, and b refer to the engine, circulating pump, air pump, feed pump, and boiler, respectively. Then the steam consumption is

$$(w_e + w_c + w_a + w_f) \div \text{horsepower or kilowatts output.}$$

To determine the heat consumption, the various values of H should be used for strict accuracy, as determined by the pressure and quality of the steam just before it is delivered to each unit, but in most cases the following form is practically correct.

$$\text{Heat consumed per hour} = w_e(H_e - h) + (w_c + w_a + w_f)(H_b - h).$$

Dividing this by the horsepower or kilowatts gives the required quantity.

The heat consumed per hour by the engine and auxiliaries may also be determined by subtracting the heat lost to leakage and drips from the total heat added to the feed water. (See (b) and (c) following.)

The values of H should be obtained from the steam tables by reference to the measurements of pressure, temperature, and quality of the steam at the point considered. For H_b this should be immediately beyond the

main stop valve of the boiler; for H_e, it should be just behind the main stop valve of the engine, between it and the separator if one is used. h is obtained from the temperature of the feed water just before it enters the boiler and obtaining enthalpy from the steam tables. If an injector is used, the temperature should be taken before entering the injector, since the heat added by that instrument comes from the boiler and returns to it. The heat consumed by the feed pump in this case should be figured separately and is very nearly equal to the heat equivalent of the work done in pumping.

(b) Heat Added per Hour to the Feed Water. If there is only one source of supply yielding W pounds per hour, then the heat added is

$$W(H_b - h).$$

If, in addition to this, there is another supply such as would be obtained from traps, separators, etc., draining back to the boiler in an independent pipe carrying W_2 pounds per hour, then the heat added is

$$W_1(H_b - h_1) + W_2(H_b - h_2),$$

the subscripts 1 and 2 referring to the main and auxiliary supplies, respectively.

The quantities H_b and h are measured as under (a). The weight of the main supply is generally measured by a weighing system as for boiler trials. (See Test 60(b).) Auxiliary supplies of feed may be metered or directly weighed during a special test made for that purpose.

(c) Heat Lost to Leakage and Drains. Leakage may occur from the steam space of the boiler or from the steam pipes either to the air or to pipe branches, and it may occur below the water level in the boiler. Leakage which cannot be prevented should be measured by a preliminary test similar to that described in Test 51. It is assumed that the rate of leakage thus determined remains the same throughout the test of the plant. Calling it w_l pounds per hour, the heat lost is

$$w_l(H_b - h).$$

If the separators, traps, etc., do not drain back to the boiler, they may be kept closed during the test and drained at regular intervals into buckets. of cold water whereby they are weighed. If w_d is the number of pounds of water withdrawn per hour, the loss is

$$w_d(H_b - h).$$

If this water is returned as an auxiliary supply, h_2 should be substituted for h in the above.

(d) Heat Balance. All the items, having been measured as above in heat units per hour, may now be expressed as percentages of the heat added to the feed water. Then the heat balance is

100 = per cent of heat consumed by engine
    + per cent of heat consumed by auxiliaries (stated separately or col-
        lectively)
    + per cent of heat lost to leakage and drains.

The first item on the right-hand side may be subdivided into the per cent of heat equivalent to the useful work and the per cents representing the various engine losses.

It should be noted that this method of analyzing the heat distribution charges against the engine and auxiliaries the losses of radiation between the condenser and the boiler. Also, if a feed-water heater is used, the heat consumed by the engine is less, although the actual performance of the engine is exactly the same as if no heater were used, because the value of h, the heat of the feed water, is higher. The only way in which the effect of the heater appears numerically is in the heat added to the feed water, but no idea of the saving due to the heater may be formed.

SECTION V

TESTS OF REFRIGERATING MACHINERY

77. TEST OF A COMPRESSION MACHINE

Principles. The student, it is assumed, is informed on the mechanical and thermal features of the refrigerating machine, descriptions of which are available in numerous treatises.

Following are given definitions of the quantities usually sought in tests of refrigerating machinery.

Refrigerating effect is the amount of heat abstracted by the refrigerant from the cooling medium, such as brine, expressed in B.t.u. per unit of time (minute, hour or 24-hr. day). This includes the waste-cooling effect due to heat transferred from bodies it is not desired to cool—a waste necessarily ensuing from imperfect insulation, manipulation, etc.

The unit of refrigeration is a refrigerating effect of 288,000 B.t.u. per 24 hr., or 200 B.t.u. per min. To obtain this effect by ice at 32° F., it would be necessary to melt 1 ton (2000 lb.) of it in 24 hr., the latent heat of fusion of ice being 144 B.t.u. Units of refrigeration are therefore spoken of as *tons* more commonly than as B.t.u.

The capacity of a plant equals the number of units of refrigeration it delivers: That is, the number of tons of 32° ice it would be necessary to melt per 24-hr. day to produce a refrigerating effect equivalent to that of the ammonia. Hence, this quantity is often referred to as "ice-melting capacity."

Ice-making capacity is sometimes considered in the case of plants devoted exclusively to the manufacture of ice. Expressed in tons per 24 hr., this is equal to between one-half and eight-tenths of the ice-melting capacity previously defined, the diminution being due to the losses in the process of heat abstraction by the brine, manipulations of cans, and to the fact that the water must first be cooled to 32° and, after freezing, be chilled below that point. Rating in terms of ice-making capacity is therefore only definite when the temperatures of water and finished ice are stated.

The coefficient of performance is a more correct term for what is sometimes miscalled the efficiency of a refrigerating system. This quantity is the ratio of the refrigerating effect in B.t.u. per unit of time to the heat equivalent to the indicated work done by the compressor in the same time; that is, it is the useful effect divided by the power put in, and in this respect is superficially an efficiency. The term efficiency generally has reference to the relation of an energy, after passing through a conversion, to its value at the source. In the case of refrigerating machinery the compressor energy is *not the source* of the useful effect. In reality, it is the condensing water that does the cooling, the compressor being merely an auxiliary in the process. As the work done upon the refrigerant is the paid-for item and as the refrigerating effect is the thing desired, it is useful to know the ratio of the two.

The ideal coefficient of performance is the maximum that could be obtained under a given set of operating conditions if there were no losses. It is fixed by the operating temperatures of condenser and refrigerator and is equal to

$$P_i = \frac{460 + T_r}{T_c - T_r},$$

in which the numerator is the absolute temperature of the refrigerator; T_r is its temperature in degrees Fahrenheit, and T_c is the temperature of the condenser in the same units. It should be noticed that neither refrigerator nor condenser is ever at one uniform temperature in operation. To maintain the heat flow, the brine must always be a little warmer than the refrigerant it boils, and the circulating water always a little cooler than the vapor it condenses. But, assuming ideal apparatus, the vapor could be worked in the refrigerator up to the temperature of the outgoing brine (just as in a perfect steam boiler the steam might be worked up to the temperature of the outgoing flue gases), and in the condenser the vapor could be worked down to the temperature of the outgoing water. If these temperatures be used in the calculation, then, the resulting coefficient of performance represents ideal conditions of the whole system, including heat transmission of refrigerator and condenser. If, however, the temperatures corresponding to the vapor pressures of the refrigerant are used, the resulting coefficient is that of the system, exclusive of losses in heat transmission in refrigerator and condenser. Numerical examples of these two values of the ideal coefficient of performance will be cited later.

The efficiency of the plant may be expressed as the ratio of the actual to the ideal coefficient of performance.

The economy of compression plants is often stated in pounds of refrigerating effect (ice melting) per pound of coal, on the assumption that a certain number of pounds of coal are consumed to make one indicated horsepower at the compressor.

Other results should include the expense of condensing water (under certain conditions this may be as large as the cost of the fuel) and the cost of auxiliary power.

For complete tests, enabling a study of all the heat transfers, temperature and time-quantity determination should be made of both ammonia and cooling water.

The duration of the trial should be at least 12 hr., and preferably 24, to allow for the possible error at the finish of the test due to a different amount of heat being stored in the refrigerator, condenser, etc., from that contained at the beginning.

(a) **Refrigerating Effect by Measurement of the Brine.** Refrigerating effect in B.t.u. per minute, equals

$$R = W(T - t)C$$

in which W = weight of brine, in pounds per minute;

$\quad\quad T$ = temperature of brine at inlet, in degrees F.;

$\quad\quad t$ = temperature of brine at outlet, in degrees F.;

$\quad\quad C$ = specific heat of brine.

For the determination of W the various methods of water measurement have been applied. Owing to the large quantity of brine circulated even in small plants, the method of direct weighing by tanks and scales is inconvenient. A modification of this consists in running the brine through two tanks, one above the other, the upper one having in its bottom a number of orifices through which the brine flows. The rate of flow varies with the height of the brine level in the upper tank. One of the orifices may be readily calibrated during the trial by collecting, weighing and timing its discharge at various heads. The total brine flowing through all the orifices may then be found by multiplication.

When meters calibrated in gallons or cubic feet are used, a separate determination of the weight of the brine per gallon or cubic foot is necessary. This may be made with a hydrometer, care being taken to test a sample withdrawn near the meter and at approximately the same temperature as it has in the pipe.

Concerning temperatures T and t, as the range is only between 5 and 10°, finely graduated thermometers must be used. For not more than 2 per cent of error the graduations must be about 2 per cent of the temperature range—that is, between one-fifth and one-tenth degree. The thermometers should be inserted in wells filled with mercury and should be located in the inlet and outlet pipes as near as possible to the cooler. The protruding portions of the wells should be well insulated. During the trial it is well to interchange the thermometers as a check on their accuracy.

The specific heat of brine varies with its concentration, constituents and temperature. The last-named variation is comparatively small—about 0.05 per cent decrease of the specific heat for each degree decrease in temperature. As to the effect of the constituents, the presence in calcium chloride brine of manganese and sodium chloride in moderate quantities (say, up to 20 per cent) does not affect the specific heat of the mixture materially. The effect of the concentration, however, is to lower the specific heat from unity (that of water) down to about 0.65 at a concentration corresponding to a specific gravity of 1.26.

The following formula for pure calcium chloride brine will give results for the specific heat of commercial calcium chloride brine to within 1 or 2 per cent of error, between the limits of −4 and 40° F., and specific gravities between 1.10 and 1.26:

$$C = 1.833 - 0.93G - 0.0005(32 - T_a).$$

In this formula * G is the specific gravity, and T_a is the average temperature of the brine in degrees Fahrenheit at inlet and outlet.

(b) Refrigerating Effect by Measurements of Refrigerant. The heat added to the ammonia in the brine cooler equals the heat lost by the brine. If the former quantity be found it is therefore a measure of the refrigerating effect, and we can put

$$R = A \times (H - h'),$$

in which R is as before, A is the number of pounds of refrigerant circulated per minute, and

$H =$ the total enthalpy of the refrigerant per pound leaving the cooler;
$h' =$ the enthalpy of the liquid at the expansion valve.

To find A, see Test 78 (h).

* Deduced from the values given in Bureau of Standards Bulletin No. 135.

H and h' are to be found from tables of the properties of the refrigerant used, or from the Mollier diagram. To use these tables, the same data as for steam are necessary. For H the suction pressure must be read, together with the temperature, so that the heat of superheat may be calculated.

It often is difficult to get the exact amount of superheat of the refrigerant leaving the cooler, and also to maintain a uniform quantity of refrigerant in the cooler to avoid the error of starting and stopping as in a boiler test. For these reasons the refrigerating effect as obtained by this method may be expected only roughly to check the result by the brine method which is therefore to be preferred.

When the cooling is done by the direct expansion of the refrigerant, there is no choice between these methods, as brine measurements are eliminated under such conditions. An approximation of the refrigerating effect can be made by still another method which is also useful as a rough check of either (a) or (b).

(c) **Approximate Calculation of Refrigerating Effect.** Neglecting effects of radiation, etc., the heat added in the refrigerating plant equals the heat equivalent to the compressor work, plus the heat added to the ammonia, R. This sum equals the heat taken away, or the heat imparted to circulating water. By finding the heat taken up by the condenser water and jackets and subtracting from this the heat equivalent to the compressor work, we thus have a rough measure of the refrigerating effect.

The condenser water may be measured as under (g). Its temperature rise should be obtained by thermometers at inlet and outlet, from which data the heat removed may be calculated.

(d) **Ice-melting capacity** is readily calculated from the value of the refrigerating effect R in B.t.u. per minute, by dividing by 200, the result being in tons per 24 hr.

(e) **To obtain the actual coefficient of performance,** it is necessary to indicate the compressor. The process here is similar to that for steam-engine trials and hardly needs comment other than that a special steel-lined indicator must be used with as short pipe connections as possible to avoid materially increasing the clearance space. Having obtained the indicated horsepower of the compressor, the coefficient sought is

$$P_a = \frac{R}{42.4 \times \text{I.hp.}}.$$

(f) **The ideal coefficient of performance** may be found from the average thermometer readings at the brine and condenser water outlets, as

previously defined. Or, if vapor pressures are used, the corresponding temperatures may be found from the tables.

(g) Gallons of Cooling Water per Minute per Ton. The measurement of the condenser water is simpler than that of the brine, the quantity being much less. Direct weighing or the usual forms of water meter may be employed, care being taken to select one appropriate to the conditions of flow, whether pulsating or steady. The number of gallons of cooling water per minute per ton of ice-melting capacity should be figured for purposes of comparison and also in order to add the cost of this item to the other costs to obtain the total. This item should include all cooling water used, such as compressor jacket water and water to steam condenser if one is used.

(h) Over-All Economy. The performance of auxiliaries may be either estimated or measured. When they and the compressor are steam driven, it is well to measure the total steam supplied to the plant by the boiler-feed method. The steam consumption of the compressor and of the whole plant may then be stated in pounds of steam per ton of ice-melting.

Ratings are generally based on head and suction pressures of 185 and 15.3 psi. gage, respectively, corresponding to temperatures of 96 and 0° F., respectively. Any departure from these conditions should be considered in making comparisons of test results.

78. Test of an Absorption Machine

(Ammonia Absorption System)

Principles. In some respects the results from and methods for the test of an absorption plant are identical to those applying to the compression system, for which see Test 77. The capacity in terms of ice-melting effect may be found in the same way, and the investigation of the condenser operation is the same. The actual coefficient of performance may not readily be obtained, since the energy imparted to the ammonia (barring the work of the ammonia pump, which is comparatively small) is in the form of heat instead of compressor work. But the ideal coefficient of performance may be determined as for a compression plant and may be useful for purposes of comparison of the two systems.

The principal economy result is the steam consumption expressed in pounds of steam per hour per ton of refrigerating effect. This may be determined *in toto* for the plant or itemized as live steam to the pump and steam to the generator, the latter being subdivided into live and

exhaust steam according to whether the one or the other or a combination is used. Under favorable circumstances the generator will use 3C lb. per hr. per ton of steam. The steam consumption of ammonia pumps is subject to wide variations due to their construction. Although the energy delivered is small compared with the heat transferred in the generator, a test of this pump should be included in economy trials because of the bearing of its performance on the steam consumption of the plant; and the pump test should preferably include a determination of the horsepower delivered.

In addition to these results it is desirable to ascertain the condition of the aqua ammonia and, if convenient, the amount of anhydrous circulated per unit of time. The former is expressed in terms of its concentration; that is, the part of a pound of ammonia in one pound of the aqua ammonia (more frequently expressed as a percentage). The concentration results show the working of the absorber and generator and enable calculations regarding the ammonia circulation.

(a) **Refrigerating effect,** 2 methods. See Test 77 (a) and (b).

(b) **Ice-Melting Capacity.** See Test 77 (d).

(c) **Ideal Coefficient of Performance.** See Test 77 (f).

(d) **Gallons of Cooling Water per Minute per Ton.** See Test 77 (g) in so far as condenser water is concerned. All other cooling water supplied, if in separate amounts, should be included in this total, by similar methods of measurement.

(e) **The steam consumed by the generator** is readily ascertained by piping the generator-trap discharge into a weighing barrel. The barrel, to start with, should be about half full of cold water to prevent evaporation of the condensate from the trap. Quick emptying should be provided for. Weights should be recorded at uniform time intervals (to show uniformity of operation), from which data the rate in pounds per hour may be had.

Quality determinations of the steam are important, unless an efficient separator is installed to remove the moisture from the incoming steam, to the consideration of the results, as also are pressure readings.

If reduced live steam is used in addition to exhaust and it is sought to measure them separately, a steam meter may be applied to the live-steam line. The exhaust-steam quantity then equals the total trap condensate minus the meter reading. Measurement of the feed water, as for a boiler test, may also be resorted to, in cases where the plant may be isolated.

When the generator is supplied with exhaust steam from the pumps plus reduced live steam, the live steam may be found by subtracting from the generator trap discharge the steam consumed by the pumps. This last may be measured as follows:

(f) Steam Used by Ammonia and Brine Pumps. A separate test should be made to determine this, but if conditions are not favorable, the steam required to drive the pumps may be estimated from the manufacturer's figures for their water rates and from the horsepower actually delivered. These remarks of course apply to steam pumps; where power pumps are installed, the energy may be considered in the list of costs.

(g) Work of the Ammonia Pump. It is preferable to find the net horsepower output rather than the indicated horsepower. From the data required for the net horsepower and from the number of pump displacements, the slip may be calculated as in ordinary pumping-engine trials.

The required formula for horsepower is

$$\text{Net horsepower} = \frac{144(P_1 - P_2)V}{33,000},$$

in which P_1 and P_2 are the gage pressures worked against in pounds per square inch (that is, of the generator and absorber, respectively) and V is the number of cubic feet of strong aqua circulated per minute.

The quantity of strong aqua V may be determined in several ways. A specially calibrated meter may be used on the discharge side of the pump; the aqua may be measured in calibrated tanks on the suction side; or it may be calculated from the amount of anhydrous circulating or from the quantity of weak aqua, the concentrations being known. For this last method (see (j)).

When a receiver is installed to supply the ammonia pump, it may be readily calibrated so that the volume contained, corresponding to any height of the liquid in the gage-glass, is known. Now, by cutting off the flow from the absorber into this receiver for a short interval, the rate, in cubic feet per minute, at which the strong aqua is pumped out, may be ascertained. If desired, two receivers may be arranged in parallel, thereby enabling continuous measurement.

(h) Weight of Anhydrous NH$_3$ Circulated per Minute. Volume measurements may be made in the same way as described under (g) for strong aqua. To convert the volumetric results into weights, a temperature or pressure reading also should be obtained to find the density of the liquid from the tables of properties of ammonia. When closely accurate results on the anhydrous quantity rate are desired, a favorite method is by

direct weighing. Two ammonia drums, arranged in parallel, are piped in the line between the condenser and expansion valve, so that they can be filled and emptied alternately. These drums are set on platform scales, the connections to them being horizontal and sufficiently long to deflect about ¼ in. under a load at the end of about ¼ lb. The drums and contents may then be weighed without sensible error. The increase of accuracy of this procedure probably does not justify the elaboration of the apparatus.

Suitable calibrated fluid meters may be used to measure the weight of ammonia circulated or calibrated receiving tanks, placed in parallel, may also be used. In the latter case, each of the tanks is fitted with a gage glass to indicate the height of the liquid in the tanks. Valves are installed so that one tank can be accumulating liquid while liquid is being supplied to the system from the other. This method, of course, measures the volume of the liquid from which the weight may be calculated by reference to the table of properties of liquid ammonia.

It is useful to find the number of pounds of anhydrous ammonia per minute per ton of refrigerating effect and to compare this value with the amount theoretically required as tabulated in handbooks or calculated from the heat contents under the prevailing conditions.

(i) Concentration and Specific Gravity of the Aqua. The concentration of aqua ammonia is determinable from its specific gravity. Given a definite solution of ammonia in water, its specific gravity is fixed at any one temperature, say, 60°. At any higher temperature the solution occupies more volume, owing to expansion, and therefore has a lower specific gravity, although the concentration remains the same. Or to put it another way, for any value of the specific gravity the concentration depends on the temperature.

The chart, Fig. 114, gives values of the concentration of 60° aqua, corresponding to various specific gravities.

Because of the volatility of aqua ammonia at the high temperature, and reduced pressure prevailing when a sample is drawn, considerable care must be taken when testing for concentration. The following procedure is recommended. Outlets for samples of the aqua are arranged at points where the temperatures are comparatively low; that is, on the discharge side of the pump for the strong aqua and between the absorber and weak-aqua cooler for the weak. These outlets should be fitted with short lengths of rubber tubing. A glass graduate, reading preferably in cubic centimeters, is provided, together with a hydrometer or a sensitive

scales. The graduate is about half filled with water for the first trial
(for the second, a somewhat different amount may be selected, depending
upon the outcome of the first) and then placed in a bucket of cold brine
until it and its contents are chilled to about 32°. After running a little
aqua through the sampling tube, it is directed below the surface of the
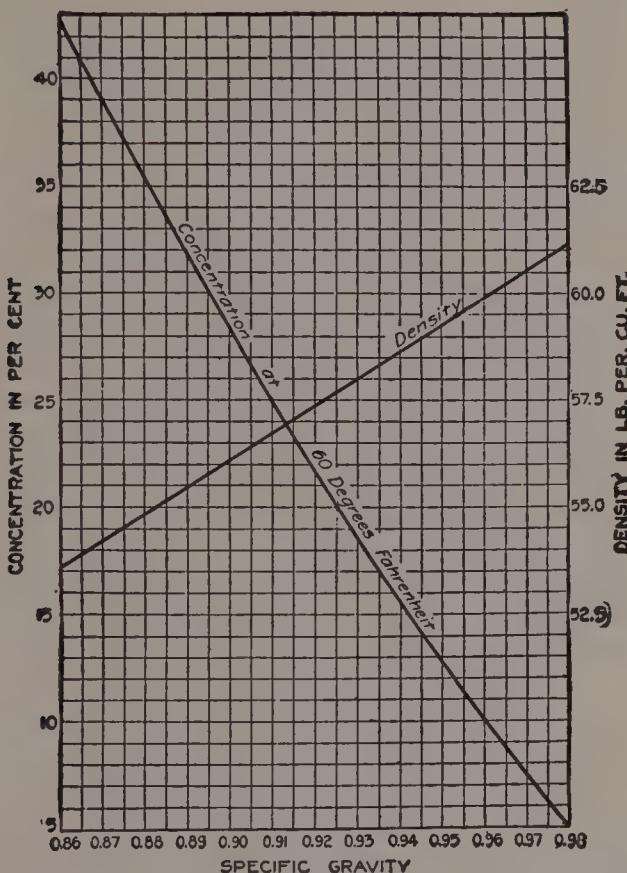
Fig. 114.—Concentration of Aqua Ammonia at 60° F., and Density Variation.

water in the graduate until the mixture of cold water and incoming aqua
attains a temperature of about 60° as shown by a thermometer. The
specific gravity of the mixture should now be taken quickly with a
hydrometer or by weighing the contents of the graduate and then by
calculation. Provided the temperature is within 5 to 10° of 60°, the
concentration of the mixture may now be found on the chart, Fig. 114,
against the determined value of its specific gravity. It is the concen-
tration of the sample that is sought, however, and this may be calculated
from the following relation:

$$\text{Conc. of sample} = \text{conc. of mixture} \div \left(1 - \frac{R}{S_m}\right),$$

in which R is the ratio of the volume of cold water, before mixing with the sample, to the volume of the mixture, and S_m represents the specific gravity of the mixture.

If scales instead of hydrometer are used to obtain the weights of cold water and mixture, then the concentration of the sample is more readily figured from the relation

$$\text{Conc. of sample} = \text{conc. of mixture} \times \frac{\text{weight of mixture}}{\text{weight of sample}}.$$

If the temperature of aqua ammonia is higher than 60°, its specific gravity will be lower by between 0.001 and 0.005 for each 10°, depending upon the concentration, the stronger solutions having a higher coefficient of expansion and therefore needing greater correction than the weaker. Under these circumstances the specific gravity S_{60} of a given solution at 60° can be figured from its specific gravity S_t at $t°$, the following formula, which gives moderately accurate results, being used: *

$$S_{60} = S_t + 0.003(t - 60)(1 - S_t).$$

For example, if the specific gravity is found to be 0.865 at 75°, then at 60° it equals

$$S_{60} = 0.865 + 0.003(75 - 60)(1 - 0.865) = 0.871.$$

The value 0.871 is then to be referred to the chart for the corresponding concentration.

When the reverse calculation is sought (for the density determination of aqua at high temperatures) this formula may be expressed thus:

$$S_t = \frac{S_{60} - 0.003(t - 60)}{1 - 0.003(t - 60)}.$$

(j) Calculation of Aqua and Anhydrous Weights per Minute from the Concentrations. This method is based upon the following rational relations:

$$A_s = \frac{1 - C_w}{C_s - C_w} \times A,$$

---

* Curves presented in Marks' Mechanical Engineers' Handbook may also be used See also the equation and chart given under Test 79(a).

$$A_s = \frac{1 - C_w}{1 - C_s} \times A_w,$$

$$A_w = \frac{1 - C_s}{1 - C_w} \times A_s,$$

$$A = A_s - A_w,$$

in which A_s, A_w and A are the weights in pounds per minute of the strong and weak aqua and anhydrous respectively, and C_s and C_w are the con-
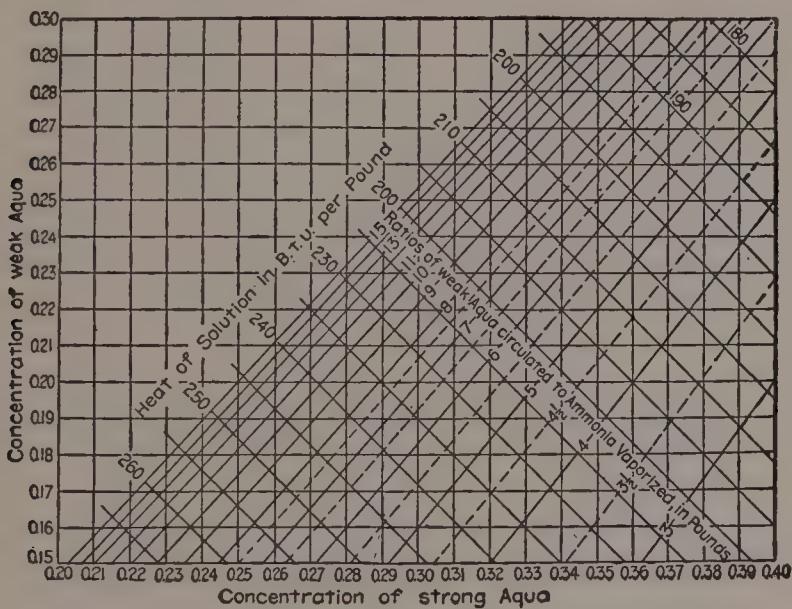
Fig. 115.—Heat of Solution of Aqua Ammonia.

centrations of the strong and weak aqua, respectively, in pounds of NH_3 per pound of solution.

For direct measurement of these quantities, much the same methods may be employed as for the strong aqua. (See (g).) The reader is reminded that only one of the three quantities, A, A_w, or A_s, need be directly measured for a fairly accurate knowledge of the performance of the plant when the concentrations are known. The ammonia, whether aqua or anhydrous, may therefore be directly metered at a point selected according to the existing layout. It should be borne in mind, however, that the results depending upon concentration figures may not be exact, because of the difficulty in getting close values of the concentrations.

It is to be noted that, since these relations are between weights, a knowledge of densities is necessary to convert them into or from volumes.

For the aqua this involves a measurement of specific gravity; for the anhydrous the ammonia tables may be referred to. The chart, Fig. 115, gives values of the density of aqua corresponding to those of specific gravity, the density scale being on the right.

As an example of how these relations are to be applied, let us suppose that the concentrations of the strong and weak aqua are 38 and 26 per cent, respectively (the concentrations having been found as under (i)), and that the number of pounds of anhydrous circulated per minute is 20. Then,

$$A_s = \frac{1 - 0.26}{0.38 - 0.26} \times 20 = 123 \text{ lb. per min.}$$

Referring to the chart, it is seen that 38 per cent aqua has a specific gravity of 0.873, if the aqua is at 60° F., and a density of 54.4 lb. per cu. ft.

For the calculation of (g), if the temperature of the aqua at the pump is not much higher than this, then the cubic feet discharged per minute is $123 \div 54.4 = 2.26$, which is the value of V to be used in the formula for horsepower.

79. Heat Balance of a Refrigeration Plant

(Ammonia Absorption System)

Principles. The over-all heat transfers in an absorption refrigeration plant may be summarized very briefly as follows: Energy is added to the system, first by the steam supplied to the generator, second by the work of the ammonia pump, and third by the heat added to the brine during the refrigeration process. Heat is removed from the system by circulating water, first in the ammonia condenser, second in the absorber, and lastly, in the weak-aqua cooler and the rectifier. The energy added equals the heat removed, as itemized, plus or minus radiation.

A study of this over-all heat balance is useful, but before making comparisons of grand totals, it is preferable to analyze the performance of separate units. Of these it is the purpose here to deal with the generator and the absorber.

To review the action of the generator, strong aqua ammonia enters at a comparatively low temperature, to which heat is added by steam pipes. Some of the ammonia is thereby boiled out of the water holding it in solution. There results from the process a quantity of superheated anhydrous ammonia, mixed with a small percentage of steam, and a larger

quantity of weak aqua, comparatively hot, which is to be returned to the absorber. The heat in the steam supplied goes to vaporize and superheat ammonia from the form of strong aqua and to raise the temperature of the entering aqua to a higher value upon leaving. Furthermore, a certain quantity of heat is required to break the bond between the liquid ammonia and the water holding it in solution, before vaporization can take place. This quantity is additional to that required to vaporize liquid anhydrous ammonia and is referred to as the **"heat of solution."** Experimental values of it have been made showing that it is independent of pressure and temperature and varies only with the concentration of the solution, being about 347 B.t.u. per lb. for very weak solutions (approximately to zero percentage of ammonia) and diminishing as the concentration is increased to a zero value when the concentration is 60 per cent.

It is convenient to base the heat calculations of the ammonia upon a single pound of the NH_3 going through a complete cycle of temperature and state changes. From these results hourly quantities may be figured by multiplying by the number of pounds of anhydrous circulated per hour. Thus, let A represent the latter quantity and W the number of pounds of weak aqua circulated per pound of anhydrous vaporized; then the heat transferred in the generator per hour $= A \times (W \times$ heat added to 1 lb. of the aqua $+$ heat of solution in B.t.u. per pound of $NH_3 +$ difference in "heat content" of NH_3 per pound, from tables). The quantity in the parentheses is the heat transferred for the circulation of one pound of the anhydrous.

The product, representing B.t.u. per hour, must equal the heat given up by the steam supplied to the generator, if the losses due to radiation and steam in the superheated ammonia are ignored.

The measurements necessary to be made for the determination of this heat balance are in part the same as those for an economy trial, for which see Test 68. In addition, it is necessary to determine temperatures of the ingoing and outgoing aqua near the generator, the temperature of the ammonia vapor before it enters the rectifier and the head pressure— readings of which should be taken sufficiently to insure fair averages. The vapor data are used to determine the "heat content" from the tables (see Appendix), or Mollier diagram for ammonia. For the steam-heat quantities, pressure and quality must be ascertained as well as the temperature of the condensate at the generator trap.

(a) **To determine the number of pounds of weak aqua circulated per pound of anhydrous** (that is, W), it is necessary to measure the concentrations of the weak and strong aqua (referred to as C_w and C_s, re-

spectively) as described under Test 78(i). The desired quantity may then be calculated from the rational formula,

$$W = \frac{A_w}{A} = \frac{1 - C_s}{C_s - C_w}.$$

A simpler procedure may be adopted through the use of the accompanying curves. Referring to Fig. 115, the lines slanting diagonally upward from left to right show values of the number of pounds of weak aqua per pound of anhydrous corresponding to various combinations of concentrations. To use these curves assume, for example, that the concentration of the strong aqua is 0.34, and of the weak, 0.24. On the chart the lines representing these values intersect between the curves marked 6.5 and 7, at about one-fifth of the distance across, so the value of W is 6.6.

This chart also gives the "heat of solution," which is plotted from the experimental values of H. Mollier. As previously remarked, this depends only upon the concentration. During the driving-off process in the generator, the concentration varies from that of the strong aqua to that of the weak. It is the average concentration, then, that determines the heat of solution. This condition is met in the chart, as before, by the intersection of the lines representing the concentrations; the location of this point with relation to the "heat lines" (slanting downward from left to right) gives the heat of solution. For example, using the concentrations 0.34 and 0.24, the point of intersection is about four-fifths of the distance across from the 205 to the 210 B.t.u. line. Therefore the heat of solution is 209 B.t.u.

(b) Heat Added to NH_3 and Aqua in Generator. We now have means of finding the number of pounds of weak aqua per pound of ammonia vaporized and the heat of solution per pound. Referring to the heat equation for the generator, already given, it will be noticed that there are still to be found the heat added to 1 lb. of the weak aqua and the difference in "heat content" of the NH_3 before and after vaporization. The latter is readily found from the ammonia tables or Mollier diagram, being the difference between the total heat of the NH_3 vapor at the head pressure and outgoing temperature and the heat of the liquid NH_3 at the temperature of the incoming strong aqua.

Considering, now, the heat added to 1 lb. of the aqua passing through the generator, it is to be remarked that there are no tabulated values of the heat of the liquid of mixtures of ammonia and water, or of the specific heats of such mixtures. It will, however, be sufficiently accurate to

assume that the heat capacity of a combination of ammonia and water in definite proportion is the sum of the heat capacities of the constituents, according to their proportion. Thus, if we call the heat of the liquid of ammonia h_a and of the water h_w (as tabulated in the ammonia and steam tables, respectively), then for a mixture of C pounds of ammonia
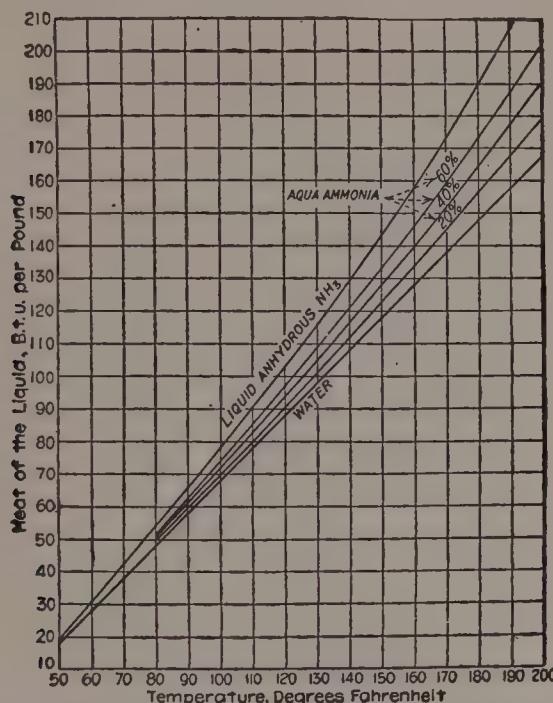
FIG. 116.—Heat of the Liquid for Water, Aqua Ammonia and Anhydrous Ammonia.

and $1 - C$ lb. of water, the heat of the liquid (aqua ammonia) $= h_a C + h_w (1 - C)$. Values of this quantity at various temperatures and concentrations are shown graphically in Fig. 116, the highest curve of which gives (vertically) the heat of the liquid of pure ammonia, the lowest curve, of unmixed water, and the intermediate curves, of mixtures of the two denoted by their concentrations. The heat of the liquid of aqua at concentrations intermediate between those shown may be obtained with sufficient accuracy by interpolation.

Coming now to detailed calculations, let us assume the following data:

Concentration, weak aqua, $C_w = 0.24$;
Concentration, strong aqua, $C_s = 0.34$;
Head pressure $= 100.3$ lb., gage;
Temperature of outgoing $NH_3 = 150°$ F.;
Temperature weak aqua $= 180°$ F.;
Temperature strong aqua $= 130°$ F.

From Fig. 115 is obtained the value 6.6 as the number of pounds of weak aqua leaving the generator per pound of ammonia vaporized. From Fig. 116 the heat of the liquid of 1 lb. of 24 per cent aqua is found to be 158 B.t.u. at 180° and 102 B.t.u. at 130°. Consequently, the heat added to the aqua per pound of ammonia vaporized is $6.6 \times (158 - 102) = 370$ B.t.u. From Fig. 115, from the given concentrations, heat of solution = 209 B.t.u. From the ammonia tables, the total heat of ammonia at 100.3-lb. gage and 150° is 609 B.t.u., and the heat of liquid ammonia at 130° (its incoming temperature) is 115 B.t.u. (also obtainable from Fig. 116). The heat added to the NH_3 to raise it from the liquid to the superheated condition is consequently $609 - 115 = 494$ B.t.u. Adding these three heat quantities ($370 + 209 + 494 = 1073$ B.t.u.), we have the heat added in the generator to effect the complete circulation through the plant of 1 lb. of anhydrous ammonia. As before mentioned, this ignores the loss through vaporization of water with the ammonia, afterward removed by the rectifier, but this loss should be small.

(c) **The steam-heat quantities** involve the determination of the weight of steam used for a corresponding ammonia vaporization and may be made as for an economy trial (see Test 78 (e)). It should be noted that

$$\text{Pounds of steam per hour} = \frac{\text{heat added to } NH_3 \text{ and aqua per hour}}{\text{heat removed from 1 lb. of steam}}.$$

(d) **The heat transfers in the absorber** are essentially the same as those in the generator, the only difference being that they proceed from the ammonia media into circulating water, instead of from steam into ammonia media, and heat is given up, instead of taken in, by the NH_3 and aqua. The aqua is pumped from the absorber at a temperature lower than that at which it entered, whereas in the generator the reverse was the case. The heat of solution is *released* in the absorber and must be removed by the circulating water, also a reverse process. The anhydrous vapor, coming from the refrigerator at comparatively low pressure, is, taken into solution by the weak aqua, consequently giving up (in addition to the heat of solution) an amount equal to the total heat of the incoming anhydrous minus the heat of the liquid NH_3 at the temperature of the outgoing (strong) aqua. It will thus be seen that the measurements and calculations for the determination of the heat given up by the absorber per pound of anhydrous ammonia are exactly the same as for the generator.

(e) Heat Removed by Condenser Water. The heat calculated under (d), expressed in B.t.u. per unit of time, equals the heat removed from the circulating water in the same time, plus or minus radiation. To obtain the heat removed, it is necessary to measure the circulating water for a definite time, as for an economy trial, and its temperature range between entering and leaving the absorber (see Test 67(g)).

PART FOUR

SECTION I

MISCELLANEOUS TESTS

80. TEST OF A HYDRAULIC RAM

Principles. The hydraulic ram is a pump which uses the energy of a large volume of water under a low head in order to deliver a part of the water at an increased head. This is accomplished by establishing a flow through a waste valve at the foot of the supply pipe. When the velocity thus started attains a certain value, it closes the waste valve, whereupon the kinetic energy of the previously moving column in the supply pipe is converted into pressure great enough to open a valve into the discharge pipe. Water is then delivered at the increased pressure until the pressure energy is reduced to the static condition, when the discharge valve closes, the waste valve opens, and the cycle recommences. Fig. 117 shows diagrammatically the arrangement of piping of a hydraulic ram.
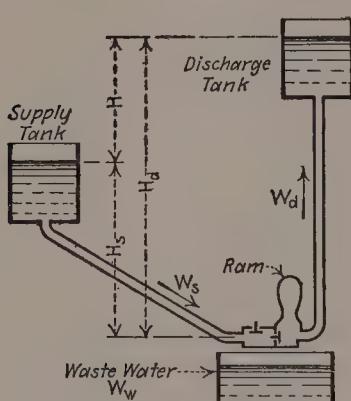
FIG. 117.—Hydraulic Ram.

Two values for the useful work of a ram may be obtained, depending upon the level from which the head pumped against is dated. Referred to the level of the supply water, the useful work is that corresponding to the elevation of a weight of water through the distance H, Fig. 117. Referred to the level of the ram, it is the work performed in lifting the same weight of water through the distance H_d. The one result leads to what is known as Rankine's efficiency, the other to D'Aubisson's. No confusion should arise through the existence of the two different efficiencies. Whether the one or the other should be used depends upon whether one is interested in pumping water from the supply level or from the level of the ram.

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Selection of the Independent Variable. Laboratory tests are generally conducted under a constant supply head, although this may be varied in the same way as for the test of a water turbine. If the supply head is maintained constant, either the number of strokes per minute of the waste valve or the discharge head may be made the independent variable.

(a) **Capacity** is expressed in gallons of water pumped per 24 hr. This may be measured by catching the discharge in a pail, or larger vessel if necessary, during a counted time. Instead of pumping against a static head as in regular operation, the discharge may be throttled by means of a valve in the discharge pipe which may then be short enough to collect conveniently the water delivered. The desired head is ascertained and regulated by the aid of a pressure gage between the ram and the throttle valve.

If the number of strokes per minute of the waste valve is the independent variable, it can be varied by changing the lift of the valve, or adjusting the spring tension, if it is operated by a spring.

(b) **Efficiencies.** Let W and H stand for the weight of water flowing per minute in pounds and head in feet, respectively; and let the subscripts s, w, and d refer to the supply, waste, and discharge, respectively, as indicated by Fig. 117. According to Rankine's efficiency a weight of water W_w must fall H_s feet in order to raise W_d pounds of water H feet Therefore,

$$\text{Rankine's efficiency} = \frac{W_d H}{W_w H_s}.$$

According to D'Aubisson's efficiency, the energy available to the ram is the denominator of the following expression, and this energy accomplishes $W_d H_d$ foot-pounds of work.

$$\text{D'Aubisson's efficiency} = \frac{W_d H_d}{W_s H_s}.$$

Since W_d was measured for the capacity determination, it is necessary only to measure the waste water to determine W_s. This may be done by allowing the waste to collect in a calibrated tank. The head H may be found from the pressure gage readings, and H_s measured before the test. It is a good plan to use a float valve in the line which feeds the supply tank so that a constant level may be maintained in it automatically.

(c) **Curves** of capacity, total amount of water supplied in gallons per 24 hr., and efficiency against the independent variable may be plotted from the data previously mentioned.

81. Test of a Hoist

Principles. A hoist is a machine by which a large weight may be lifted by the application of a small force. This is accomplished usually by passing a chain, or chains, over a number of wheels geared together in such a way that a large motion of one end of the chain downward produces a small motion of the other end upward, whereby a mechanical advantage is obtained.

The ideal mechanical advantage is the ratio of a distance moved through by the driving chain to the corresponding distance moved through by the following chain. Referring to Fig. 118, representing a hoist diagrammatically, this ratio is D/d. If there were no friction, this would be the ratio of the weight lifted, W, to the force applied F. As there is friction:
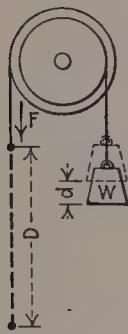
The actual mechanical advantage equals the ratio of the weight lifted to the force applied, or W/F.

The efficiency of a hoist equals the work done in lifting the weight through any distance divided by the work done by the applied force acting through the corresponding distance.

If the efficiency of a hoist is more than 50 per cent, the weight lifted will return by gravity when the hoisting force is removed unless there is a locking device. Where this is provided, it is arranged to lock against the force of gravity only, and not against a force applied to the driving chain for the purpose of lowering the weight.

Fig. 118.
Hoist.

(a) **The ideal mechanical advantage** may be determined by actual measurement of the distances, D and d, Fig. 118. As d is generally very small compared with D, the result by this method may not have the desired degree of accuracy. A better method consists of counting the number of teeth of the various gears of the hoist and figuring from the data obtained, by the principles of kinematics, the velocity ratio of driver to follower.

(b) **The actual mechanical advantage** may be determined by measuring the force applied to lift various loads. The desired results vary with the load, and may be plotted against its values. The loads may be applied by using various dead weights sufficient to cover the working range of the hoist. The force required to lift these weights may be measured by applying the force through a spring balance hooked into the driving chain. Owing to the difficulty of reading a moving instrument, this is

not a very accurate method. A better one is for the experimenter to stand on a platform scales and to apply the driving force at a uniform rate with one hand and at the same time, with the other hand, to balance the scales by adjusting the jockey weight. The weight of the experimenter minus the reading of the scales then equals the force applied to the hoist.

Several determinations of the force applied should be noted at each load, and their mean used to obtain the actual mechanical advantage at that load. It should not be attempted to make mental averages.

(c) Efficiency equals (see Fig. 118)

$$\frac{\text{Work done}}{\text{Work applied}} = \frac{W \times d}{F \times D} = \frac{\dfrac{W}{F}}{\dfrac{D}{d}} = \frac{\text{actual mechanical advantage}}{\text{ideal mechanical advantage}}.$$

Consequently each result under (b) may be divided by that under (a) to get the corresponding efficiency. The efficiencies should be plotted against the loads.

FRICTION

Friction is the force which resists the relative motion of two bodies in contact, and is due to the interference of their particles at the surface of contact. The laws controlling this force are different, depending upon whether the rubbing materials are fluid or solid. Solid friction may be either from sliding or rolling contact, but in this work sliding friction only will be considered. There is a third case, generally of greatest importance to the mechanical engineer—namely, the friction of lubricated solids. The controlling laws fall between those for solid and fluid friction, resembling the one or the other according to the amount of lubrication.

The laws of fluid friction are well established, but this is unfortunately not true of solid friction. An understanding of the latter is important in the analyses of friction drives, belt forces, etc., and important also to our knowledge of lubricated friction since the rubbing action of solids is a limiting condition of lubricated machine parts.

It has generally been assumed that the friction between two dry solids is a constant fraction of the normal force pressing the solids together, independent of the area of contact and velocity of sliding, and depending only upon the nature of the surfaces. These hypotheses have been discredited, but, thus far, experimentation upon this subject has not proceeded enough to give us more than an idea wherein they are fallacious.

It is possible that they are true for certain materials or certain conditions, but unquestionably they are not generally true.

The variation of friction, in the general case, depends upon the normal force between the surfaces, the amount of contact area, the velocity of sliding, and the materials of the bodies.

The following table is arranged to give a ready comparison of the several cases. It should be remembered, however, that the statements under solid and lubricated friction are not well established, and are possibly subject to exceptions.

VARIATION OF FRICTION

	Fluid	Solid	Lubricated Solid
Contact area.	Directly proportional.	Independent (approx.).	Directly proportional.
Velocity.	Directly proportional as the square, except at low Reynolds' numbers.	Inversely, at a diminishing rate.	Inversely for low speeds, directly as square root for high speeds.
Character of surface.	Independent.	Varies.	Varies.
Normal force.	Independent.	Directly proportional.	Independent.

It is further known that the frictional resistance to a fluid moving upon a solid is related in some way to the viscosity of the fluid, decreasing as the viscosity decreases. This property has an important bearing upon lubricated solid friction because the viscosity of lubricants varies with temperature, thus introducing another variable in this case.

For solid friction, the working hypothesis is that

$$F = fN$$

in which F is the force of friction, N the force normal to the rubbing surfaces, and f a number less than one called the "coefficient of friction."

82. BELT TESTERS

Principles. When power is transmitted from one shaft to another by means of a belt running on pulleys, friction is the useful force. Friction sets up tensions in the belt; on one side, a greater amount T_1, and on the other a lesser, T_2 (see Fig. 119). The difference between these tensions

is the net force, T, which effects a turning effort of the follower. It is shown in works on machine design that

$$\frac{T_1}{T_2} = 10^k \quad \text{and} \quad T = T_1 - T_2 = T_1\left(1 - \frac{1}{10^k}\right)$$

in which $k = 0.0076f\theta$; f being coefficient of friction and θ the angle of contact of the belt in degrees. The formulas are derived on the assumption that f varies only with the normal force of contact and that centrifugal force on the belt is negligible. For belt speeds over 2500 ft. per min., centrifugal force must be considered. Its effect is to lessen the normal force between belt and pulley, thus decreasing the effective force, and

$$T = T_1 - T_2 = \left(T_1 - \frac{12wv^2}{g}\right)\left(1 - \frac{1}{10^k}\right),$$

in which w is the weight in pounds of a strip of belt 1 in. long and full width; v, its velocity in feet per second; and g, the acceleration of gravity.

In usual operation, there is always a certain amount of "slip" between the belt and the pulleys. Instead of running at the same linear velocity of points of contact, there is a relative velocity difference between the belt and each pulley rim. This fact invalidates to some extent the formula giving the

FIG. 119.—Belt Tester.

relation between the belt tensions, since friction varies with the velocity of rubbing. Furthermore, so little is definitely known about solid friction, that the assumption $F = fN$ is of doubtful accuracy. It is very likely that, in the case of belts, the coefficient of friction f varies with the normal force N as well as with the velocity of slipping. As the formula depends upon an opposite assumption, results of the coefficient of friction obtained from it must be considered of very questionable value. On the other hand, we have no better formula, and for this reason it must be used.

Belt-testing machines are instruments for testing the efficiency of transmission, for determining the slip under given conditions, and for determining the coefficient of friction.

The efficiency of transmission may be found by measuring the power delivered to the belt with a transmission dynamometer and the power delivered to the follower pulley by an absorption dynamometer (see Fig. 119), suitable allowance being made for the friction of the shaft bearings. The difference between these quantities is the losses, which are due
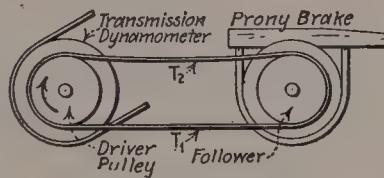
to work done by friction in moving through a distance equal to the slip and the work done in flexing the belt, windage neglected.

The slip may be ascertained by counting the revolutions per minute of the shafts; the difference between the linear velocities of the pulley rims is the total slip. The slip over either pulley may be found independently by measuring the linear velocity of the belt. This may be done by counting the number of times a chalk mark on the belt appears in a given time or by arranging a piece of metal to project from the edge of the belt so as to trip a revolution counter. Slip should be expressed in per cent of the driving pulley rim speed. Some forms of belt-testing machines have a differential gear arrangement by which the per cent of slip may be directly read.

To test for the coefficient of friction, apparatus must be provided for the determination of the belt tensions, T_1 and T_2, so that the formula may be solved for f. This is usually done by separate measurements of the sum and difference of the belt tensions; adding these results and dividing by two gives T_1, knowing which, T_2 is readily obtained.

The turning force of the belt, T, equals the difference of the tensions. The moment of this turning force equals the torque shown by the friction brake (Fig. 119). Therefore, the difference of belt tensions may be calculated by dividing the torque shown by the brake by the radius of the follower pulley. This radius should be measured to the center line of the belt.

To determine the sum of the belt tensions, not including that due to centrifugal force, one type of machine has the driver pulley mounted freely on one arm of a bell-crank lever the other arm of which bears on a platform scales. The belt pull is then indicated on the scales in inverse proportion to the lever arms. Another type of machine has the belt arranged vertically, the follower pulley being freely suspended. The sum of the belt tensions is then equal to the weight of the follower pulley and attachments together with any additional weights used for the purpose of increasing the belt tensions. In both types, the total tension may be varied.

The coefficient varies markedly with the condition of the rubbing surfaces, so every precaution should be used to keep this and other controllable conditions constant. The coefficient also varies with the humidity of the surrounding atmosphere; it is therefore well to make observations of this.

(a) **The constants of the instrument** are those of the Prony brake (see Test 6) and of any lever arms used for getting the tensions. These are readily obtainable by direct measurement.

SECTION II

TESTS OF LUBRICATING OILS

General. A number of tests have been devised which are indicative of the relative lubricating qualities of oils. There is no absolute measure of such qualities and any attempt to interpret the results of tests as absolute values of lubricating effectiveness is absurd. The tests are:

(a) Viscosity.
(b) Pour point (and cloud point).
(c) Coefficient of friction.

Still other tests have been devised which are indicative of the physical properties of oils and have little or no direct bearing on their lubricating effectiveness. The tests do, however, aid in the classification of oils and in the specification of grades with respect to physical properties. This series consists of:

(a) Specific gravity.
(b) Color.
(c) Flash and fire points.
(d) Carbon residue.
(e) Corrosion.
(f) Reaction.
(g) Neutralization number.
(h) Emulsion.
(i) Demulsibility.

The procedure involved in practically all of the tests mentioned is lengthy and complicated and requires special apparatus. They will be described very briefly, the reader being referred to the following for more detailed information on the apparatus and procedure required in each test:

1. A.S.T.M. Codes on Tests of Lubricating Oils.
2. Department of Commerce Technical Paper 323B.
3. Instruction cards accompanying the apparatus.

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83. Viscosity of Oils

Principles. Viscosity manifests itself as internal friction of the oil, a property which opposes the motion of the particles upon themselves, resulting in a reluctance to flow. It is related to, but not proportional to, the density. Since it opposes motion, it is an undesirable property, but there is such a thing as too little viscosity for good results. If an oil flows too readily, it may be squeezed out of the space between the surfaces it should lubricate, and the total amount of oil needed for proper lubrication might then become excessive. This depends largely upon the pressure between the bearing surfaces.

Instruments have been devised which measure the "absolute viscosity" of oils and other liquids. Absolute viscosity is defined as the force in dynes required to move a layer of the liquid of 1 sq. cm. area over another layer of the liquid with a velocity of 1 cm. per sec. It has been found impracticable to actually cause one plane to move over the other but the result may be obtained by application of Poisseuille's formula for capillary tubes in which the viscosity is expressed as a function of the applied pressure and the quantity discharged. The formula is,

$$\text{Abs. vis.} = \frac{\pi p r^4}{8vl}\, t,$$

where p = pressure in grams per square centimeters;
r = radius of capillary in centimeters;
l = length of tube in centimeters;
v = volume discharged in cubic centimeters;
t = time in seconds.

The unit of absolute viscosity is the centipoise.

The measurement of viscosity by the above method is not practical for most commercial lubricating oils since the time required is too great. The commercial forms of viscosimeters give a purely relative measure of viscosity. There are several standards for expressing viscosity such as the Engler, Redwood and Saybolt units (Fig. 120). In general, they are taken as the ratio of the time required for a given volume of the oil in question to flow through an orifice of a given size under a given head, or drop in head, to the time required by the same volume of another liquid, usually water, to flow under the same conditions. The three viscosimeters mentioned above are the most popular forms and are essentially the same

in construction; having a suitable container for the oil to be tested, an orifice at the bottom of the container through which the oil flows, an outer container or bath by means of which the oil in the container or tube may

Fig. 120.—Saybolt Universal Viscosimeter.

be brought to the desired temperature, and a graduated flask which measures the amount of oil discharged.

The procedure in making the test consists of first filling the container to a predetermined level; second, heating the sample to the required temperature by means of the oil-bath; third, opening the orifice and timing the discharge of the standard quantity as it flows into the graduated flask. The viscosity can either be expressed in seconds or with respect to the standard fluid.

84. Cloud and Pour Tests

Principles. The cloud point of a petroleum oil is that temperature at which paraffin wax or other solid substances begin to crystallize out or separate from solution when the oil is chilled under specified conditions. The cloud test can only be made on oils which are transparent in layers 1.5 in. thick.

The pour point of a petroleum oil is the lowest temperature at which the oil will pour or flow when chilled without disturbance under specified conditions. The pour test may be used on all petroleum oils. Both of the tests are made in the same apparatus which is made according to specifications of the A.S.T.M. It consists of an oil bottle, fitted with a cork having a hole for a special thermometer, a water-tight metal jacket fitted with a cork disc and a ring gasket, and a cooling bath in which the freezing solution may be mixed.

The procedure is as follows; the oil sample is first cleared of water and suspended solids by filtering through dry filter paper. About 2 oz. of the clear oil are placed in the bottle and the cork fitted tightly with the bulb of the thermometer resting on the bottom of the bottle. The cork disc is placed in the bottom of the jacket and the ring gasket is fitted to the bottle 1 in. above the bottom. The bottle is then inserted in the jacket and the whole set in the cooling bath in a vertical position so that the jacket projects about 1 in. from the surface of the bath.

The freezing mixtures commonly used are as follows:

    50° F., and above, ice and water;
    10° F., and above, ice and common salt;
 — 15° F., and above, crushed ice and calcium chloride;
 — 70° F., and above, solid carbon dioxide and acetone or gasoline.

At each 2° drop in temperature the sample bottle should be raised quickly and inspected for cloudiness. When such haze or cloudiness appears the reading of the thermometer, corrected if necessary, is taken as the cloud point.

The pour test is somewhat more complicated. The sample in the test jar must be heated to not over 118° F., and then cooled in air to 90° F. The sample bottle in its jacket is then placed in the cooling bath and at every 5° drop in temperature it is removed carefully and slightly tilted to detect any motion of the oil in the bottle. The temperature indicated by the thermometer when the oil ceases flowing is the pour point.

85. Coefficient of Friction

Principles. Oil friction testers are instruments for determining the coefficient of friction in the case of lubricated solids. Their operation generally depends upon the balancing of the friction by a static moment which can be measured; from this and the constants of the instrument, the coefficient may be calculated. Referring to Fig. 121, S is a revolving shaft to which is fitted a bearing carrying a pendulum. It is arranged so that the bearing swings freely and bears on the shaft with a measured force, N. The oil to be tested is fed to the bearing, and the shaft caused to revolve in a clockwise direction. The force of friction then swings the pendulum to such an angle as to establish equilibrium. If the radius of the shaft is r feet, then the moment of the fractional force, fN, is $fN \times r$. The weight W of the pendulum acts on an arm of $R \sin A$, R being the distance of the center of the weight to the center of the shaft; consequently the moment of the pendulum equals $W \times R \sin A$. As this moment equals the moment of the frictional force, we have

Fig. 121.—Oil Friction Tester.

$$fNr = WR \sin A,$$

and

$$f = \frac{WR \sin A}{Nr}.$$

For a given set of conditions, all of the quantities on the right of the equation are constant except $\sin A$, which thus becomes a measure of f.

The Thurston oil friction testing machine depends upon this principle. The force N is applied and varied by means of a spring mounted on the pendulum. By tightening this spring the two parts of the bearing may be clamped more or less tightly together. The total force, N, equals twice the tension of the spring plus the weight of the pendulum. There are a pointer and scale on the pendulum by which the total force may be read directly. Another pointer on the pendulum moves over a stationary scale graduated to values of $WR \sin A \div r$. A thermometer well is provided in the upper bearing so running temperatures may be ascertained.

The Riehlé oil friction testing machine carries a hand screw for apply-

ing the pressure, and the pressure is measured by means of a scale beam. A second scale beam measures the moment of the turning force induced by friction. By equating this moment to fNr, as in the Thurston machine, the coefficient of friction may be obtained.

(a) **Constants of the Instrument.** In the Riehlé machine, the only constant generally necessary to test is the value of r. If desired the indications of the beams may be tested according to the principles of Test 1.

With the Thurston machine, it is necessary to find WR, r, and to test the spring.

To find WR, the pendulum should be swung to a horizontal position and supported on a pedestal resting on a platform scales. A test is then made similar to that for the unbalanced weight of a Prony brake, Test 6 (a). The result should be multiplied by the horizontal distance between the point of application of the pendulum on the pedestal and the center of motion of the pendulum. This gives the desired moment, WR.

In addition to measuring the radius of the bearing, r, its length should be determined, so that the bearing pressure in pounds per square inch of projected area may be calculated.

The graduations on the stationary scale may be tested with these data by calculating their values at any values of the angle A.

To test the indications of the spring its scale should be determined according to Test 2. As the spring is a heavy one, a strength testing machine is required to compress it.

The oil to be tested should be used under the same bearing pressure and the same temperature as it will meet in service, as far as is possible. Comparative tests of different oils should be made under uniform conditions of temperature, pressure and velocity, or else the results will not be comparable.

86. Tests of Physical Properties

(a) **The specific gravity** is of interest in connection with viscosity since a high value of one generally indicates a high value of the other. The most convenient method of determination is by means of a hydrometer.

In the United States, two hydrometer scales are in general use; the Baumé, for use with liquids both heavier and lighter than water, and the American Petroleum Institute scale which largely supplants the Baumé scale for liquids lighter than water. Both are arbitrary scales whose

units are *"degrees."* The equations for conversion of degrees to specific gravity are:

Baumé $\begin{cases} \text{liquids lighter than water, Sp. Gr.} = 140 \div (130 + \text{deg. Bé.}) \\ \text{liquids heavier than water, Sp. Gr.} = 145 \div (145 - \text{deg. Bé.}) \end{cases}$

A.P.I. (liquids lighter than water only), Sp. Gr. = $141.5 \div (131.5 + \text{deg. A.P.I.})$

All at 60° F.

In the following tests, the description of the apparatus and procedures are too lengthy to be included in detail. The purpose of the tests and a few general statements concerning them will be made. For complete details, the reader is referred to the A.S.T.M. Codes or Technical Paper 323B of the Department of Commerce, Washington, D. C. This latter paper is obtained through the Superintendent of Documents.

(b) Color. The name of this test is self-explanatory. For all refined oils (naphthas, kerosenes, etc.) the Saybolt chronometer is used. The Union colorimeter is used for all lubricating oils. In both instruments, the color of the oil is compared to color standards.

(c) Flash and Fire Points. The flash point is the temperature at which the oil will vaporize fast enough to form an explosive mixture with air. This temperature should be higher than the working temperature to be encountered in service. Consequently it should be judged in connection with its service.

The fire point is the temperature at which a body of the oil will burn when a flame is placed a short distance over its surface.

In the case of lubricating oils, the open cup test is used to determine the flash point. However, if the oil shows an open cup flash point (Fig. 122) below 175° F., its flash point must then be determined by the closed cup test (Fig. 123), using either the Tag or Pensky-Martins or equivalent instruments. All fuel oils are tested in the closed cup.

(d) Carbon Residue. This test is a means of determining the amount of carbon residue remaining after an oil has been evaporated under specified conditions. The test is indicative of the carbon-forming propensities of the oil and the results must be considered in the light of other tests and the use for which the oil is intended. It is particularly useful in connection with oils to be used in internal combustion engines, domestic fuel oils, etc.

(e) Corrosion. This is a qualitative test of the corrosive properties of an oil. It consists essentially of subjecting a polished strip of copper to the action of the oil for a definite period of time and at a definite tem-
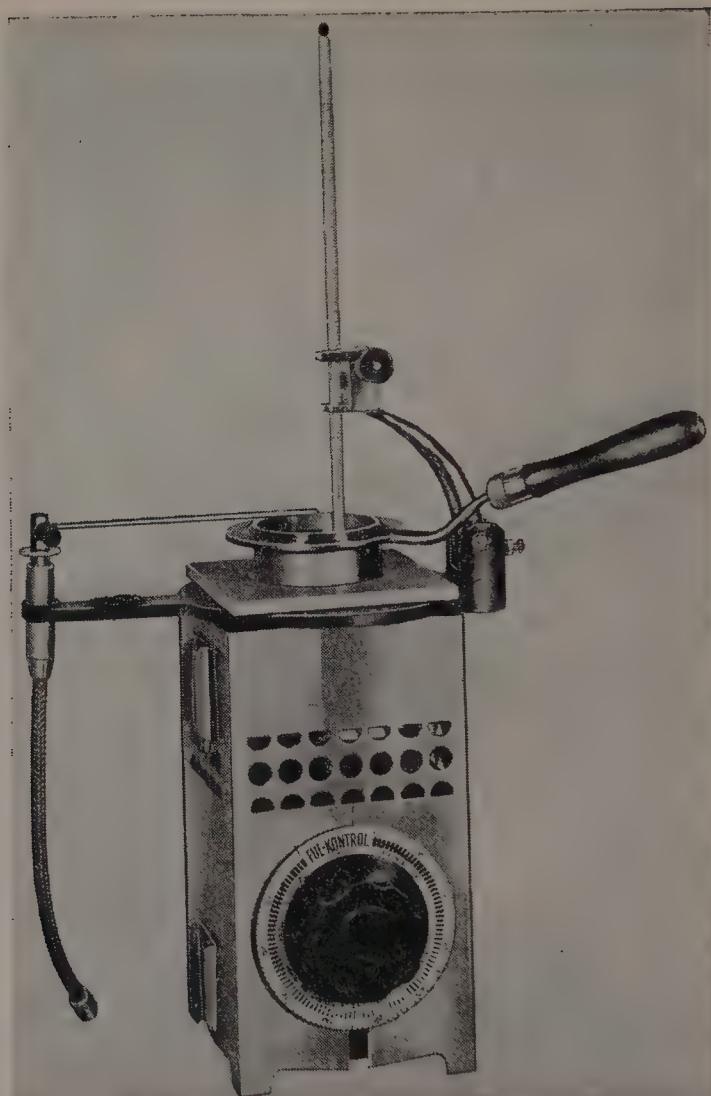
Courtesy Precision Scientific Company, Chicago

FIG. 122.—Open Cup Flash Tester.

perature and comparing the result with a freshly polished sheet of copper. The test can be applied to semi-solid products and greases as well as to the liquid products.

(f) **The reaction test** indicates the presence of alkaline substances in an oil.

(g) **The neutralization number** indicates the presence of organic constituents having acid characteristics or the contamination by alkalies and mineral acids. The neutralization number is the weight, in milligrams, of potassium hydroxide required to neutralize 1 g. of oil.

Fig. 123.—Closed Cup Flash Tester.

The alkali neutralization number is the weight of acid required to neutralize 1 g. of oil, expressed in equivalent milligrams of potassium hydroxide. The mineral acid neutralization number is the number of milligrams of potassium hydroxide required to neutralize the mineral acid content of 1 g. of oil.

(h) **Emulsion Test.** See A.S.T.M. Code.

(i) **Demulsibility.** See A.S.T.M. Code.

(j) **Endurance tests** have to do with the total amount of oil necessary to secure required lubrication. An oil that is satisfactory in all other

respects may be of prohibitive cost because its lack of body may necessitate a large rate of feed. An idea of the endurance of an oil may be formed by supplying the bearing of an oil tester with a limited amount of it and noting the time required to raise the temperature of the bearing a predetermined amount. Another method is to note the time required to raise the coefficient of friction a predetermined amount. Still another is to measure the amount of oil during a given period of time when it is fed at a rate just sufficient to prevent a rise of temperature.

SECTION III

HYGROMETRY

Hygrometry deals with the determination of the properties of mixtures of water vapor and air. Dalton's law bears directly upon this subject and should be understood.

It has been shown that a cubic foot of space at a given temperature, t, can contain no more than a fixed amount of H_2O vapor, regardless of the presence or absence of any other gas. This maximum amount is the weight of a cubic foot of saturated steam at the existing temperature, t.

Supposing that a cubic foot of space contains air at the same time that it contains the maximum amount of H_2O corresponding to the temperature, t. Then the air is said to be *saturated* and the weight of water vapor, which can be contained in a cubic foot of air under the saturated condition, is called the *capacity* of the air.

The actual amount of H_2O contained in the air may be any less than the capacity which indicated the need for means of stating the condition or *humidity* of the air. This may be done in three different ways, each having its own use.

Absolute humidity is defined as the weight of water vapor per unit volume of air; grains per cubic foot in the English system of units. It is evident that the absolute humidity will change if the unit volume of air is compressed or expanded and no water vapor is added or abstracted. This is true because, while the weight of vapor remains constant, the volume of the air is changed.

Specific humidity is a more constant property of air and has come to be used quite universally in the science of Meteorology. It is expressed as the weight of vapor per unit weight of air; grains of vapor per pound of air in engineering units.

However, when humidity is mentioned, usually relative humidity is meant. This is merely the ratio of the actual water vapor present to that at saturation. In other words, it is either the specific humidity or the absolute humidity divided by the capacity expressed in units to suit each case.

If a parcel of air, at a given relative humidity, is cooled by some means or other, the relative humidity will increase because, while the actual weight of water vapor present does not change, the capacity decreases with lowering temperature. When the temperature reaches that value where the relative humidity becomes unity, that temperature is called the dew point. Any further lowering of the temperature will cause some of the vapor to condense out in the form of minute droplets of water.

When the temperature of a given parcel of air is raised, the relative humidity must decrease.

The relative humidity is determined by the **"wet-and-dry bulb"** thermometer, or "psychrometer." This instrument consists of two thermometers, the bulb of one of which is kept wet by surrounding it with a wick saturated with water at room temperature. The evaporation from this wick lowers the temperature of the wet bulb. The dryer is the room air, the lower is the wet-bulb temperature; hence the difference between the indications of the two thermometers is a measure of the humidity. See table.

For example, if the wet bulb indicates 60° and the dry bulb 70°, then the difference is 10° and the humidity is 55 per cent.

RELATIVE HUMIDITY, PER CENT

Dry Thermometer, Degrees F.	Difference between the Dry and Wet Thermometers, Degrees F. Relative Humidity, Saturation Being 100. (Barometer = 30 in.)																										
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	26	28	30
32	89	79	69	59	49	39	30	20	11	2																	
40	92	83	75	68	60	52	45	37	29	22	15	7															
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16	10	5											
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17	13	9	5	1						
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29	25	22	19	15	12	9	6			
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38	35	32	29	26	23	20	18	12	7	
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44	41	39	36	34	31	29	26	22	17	13
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49	46	44	41	39	37	35	33	28	24	21
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57	55	52	50	48	46	44	42	40	38	34	30	26
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62	60	58	55	53	51	49	47	45	43	41	38	34	31
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66	64	62	60	58	56	54	53	51	49	47	44	41	38

Calculation of pressures and weights of air, water vapor, and mixture. The following notation will be used, all pressures in pounds per square inch absolute, and weights in pounds per cubic foot:

P_m, W_m = pressure and weight of mixture
P_a, W_a = partial pressure, and weight of the air;
P_v, W_v = partial pressure and weight of the vapor;
P', W' = pressure and weight of saturated steam at temperature, t;
t = temperature, degrees F., of the mixture;
H_r = relative humidity, per cent.

To find partial pressures, given total pressure, temperature and humidity. Find P' from the steam tables, corresponding with t.

Then

$$P_v = \frac{H_r}{100} \times P',$$

and

$$P_a = P_m - P_v.$$

The pressure of the mixture is to be found from the barometer, and the humidity as previously described.

For example, if the barometer reads 29.33″, the corresponding $P_m = 14.41$ psi. Suppose the relative humidity is 55 per cent and the temperature of the mixture is 70°. From the steam tables saturated steam at 70° has a pressure of 0.363 lb. ($= P'$). Hence, the partial pressure of the vapor in the mixture is

$$P_v = 0.55 \times 0.363 = 0.2 \text{ lb.,}$$

and

$$P_a = 14.41 - 0.2 = 14.21 \text{ lb.}$$

To find weights, given partial pressures, temperature and humidity. Find W' (density) from steam tables corresponding to t.

Then,

$$W_v = \frac{H_r}{100} \times W';$$

$$W_a = \frac{144 P_a}{53.4 \times (t + 460)} \text{ ; from } PV = RT;$$

$$W_m = W_v + W_a.$$

For example, using the previous data, the density of saturated steam at 70° is $0.00115(= W')$.

$$W_v = 0.55 \times 0.00115 = 0.00063;$$

$$W_a = \frac{144 \times 14.21}{53.4 \times 530} = 0.0725;$$

$$W_m = 0.00063 + 0.0725 = 0.0731.$$

TOTAL ENTHALPY OF AIR-STEAM MIXTURES

It was seen that W' or W_v and W_a, the weights of water vapor and of air per cubic foot of humid air, can be calculated. The enthalpy per pound of the low-pressure steam in the mixture is

$$H_f' + H_{fg}', \text{ if } 100 \text{ per cent humid,}$$

or

$$H_f + H_{fg} + C_p(t - t_s), \text{ if less than } 100 \text{ per cent humid;}$$

in which $H_f' + H_{fg}'$, in the first case, is the total enthalpy at a pressure corresponding to the temperature, t, of the mixture; and, in the second case, $H_f + H_{fg}$ corresponds to the partial pressure of the vapor, P_v, as determined in the preceding section, and t_s is the temperature of saturation of that pressure.

The general condition is that the humidity is less than 100 per cent. As the correct value of total enthalpy given above involves the use of the steam tables, etc., it is preferable to use the closely approximate formula for total heat of superheated steam at low pressures, namely,

$$H_g = 1058 + 0.455t.$$

In this connection t is the temperature, degrees F., of the steam and equal to that of the mixture.

We then have for the enthalpy of the steam in 1 cu. ft. of the mixture:

$$W'(H_f' + H_{fg}') \text{ if } 100 \text{ per cent humid,}$$

$$W_v(1058 + 0.455t) \text{ if less than } 100 \text{ per cent humid.}$$

The enthalpy of steam is the heat added to H_2O at constant pressure from 32° to bring it to a given condition. Similarly, the enthalpy of air is the heat added at constant pressure to bring it from 32° to a given condition, t. That is, the enthalpy of air is

$$C_p \times \text{weight of air} \times (t - 32).$$

Taking $C_p = 0.241$, the total enthalpy of an air-steam mixture in B.t.u. per cubic foot of the mixture is

$W'(H_f' + H_{fg}') + 0.241W_a(t - 32)$ for 100 per cent humidity,

$W_v(1058 + 0.455t) + 0.241W_a(t - 32)$ for less than 100 per cent humidity.

Heat Values in Gas Combustion. Under "Combustion of Gases" it was shown that, when complete combustion takes place, there is a contraction of volume. Thus, 100 volumes of fuel combined with 521 of air formed 80.2 volumes of CO_2, and 420.8 of N_2 (assuming all of the H_2O to be condensed upon return to standard temperature). In other words, 100 volumes of fuel + 521 of air form 501 of dry products, and the shrinkage is $501 \div 621$. This ratio is called the "coefficient of contraction."

Assume a gas requiring 4.25 cu. ft. of air to burn 1 cu. ft. of the gas. Assume, also, that 25 per cent excess air is used, and that the coefficient of contraction is 0.90. The volumes entering the combustion reaction for 1 cu. ft. of fuel are then

Fuel .... 1 cu. ft.

Air .... $4.25 + 0.25 \times 4.25 = 5.31$ cu. ft.

Products of combusion including
excess air .... $5.25 \times 0.90 + 1.06 = 5.78$ cu. ft.

Application to Junkers Calorimeter Determinations. Since, in the use of this instrument, the products of combustion, and entering air, and fuel are all at approximately the same temperature, the total heat of the perfect gases entering and leaving will undergo no change. But the air entering combustion may have any humidity, while the fuel and the products are always 100 per cent humid. Since the products contract in volume there will be a smaller volume carrying away humidity than that carrying humidity in. These two items make necessary a correction, for strict accuracy, thus:

> Correction = Enthalpy of steam in V_p cu. ft. of products,
>   − Enthalpy of steam in V_f cu. ft. of fuel,
>   − Enthalpy of steam in V_a cu. ft. of air,

which correction may be plus or minus.

This may be worked conveniently as shown below. The volumes cited above are used as an illustration and it is further assumed that temperatures of room, fuel and products are all 75°, humidity of air entering

$= 30$ per cent and of fuel and products $= 100$ per cent.   Then for the fuel and the products,

Pressure of the H_2O (at 75°) $= 0.429$ lb. per sq. in.
Weight per cu. ft. of the H_2O (at 75°) $= 0.00135$ lb. per cu. ft.
Enthalpy per lb. of the H_2O $= 1092.5$

and for the air

Pressure of the $H_2O = 0.30 \times 0.429 = 0.128$ psi.
Weight per cu. ft. $= 0.30 \times 0.00135 = 0.000405$ lb. per cu. ft.
Enthalpy per lb. $= 1058 + 0.455 \times 75 = 1092.$

Tabulating the values:

	Fuel	Air	Products
(a) Volumes in cu. ft.................	1	5.31	5.78
(b) Weight per cu. ft. of H_2O........	0.00135	0.000405	0.00135
(c) Enthalpy per lb.................	1092.5	1092.0	1092.5
(d) Enthalpy of total volumes in			
B.t.u. = (a) × (b) × (c) =	1.47	2.34	8.52

Correction $= 8.52 - 1.47 - 2.34 = 4.71$ B.t.u.

SECTION IV

ELECTRICAL MACHINERY

In mechanical engineering tests, it is frequently desirable to know the characteristics of an electric motor or generator. Motors are often used to drive various units such as centrifugal pumps and blowers. Generators are quite as frequently connected to prime movers such as steam engines and turbines. The electric dynamometer is a special form of electric generator. The following notation will be used throughout:

E = electro-motive force in volts;
I_a = armature current in amperes;
I_o = armature current, motor running free, amperes;
I_f = field current, amperes;
R_f = resistance of field, ohms.

87. Test of a Direct Current Motor

Principles. The electrical input to the motor is measured with a voltmeter and ammeter. The output of the motor can be measured with a Prony brake determining corresponding values of brake horsepower and current supplied. Such procedure is not always convenient, however, nor are the results as accurate as when the useful output is found by measuring the losses. Consequently, the latter method will be described.

The power delivered at any load is

$$746 \times \text{B.hp.} = \text{watts input} - \text{losses.}$$

The losses may be grouped into two classes: constant and variable. The **variable loss** is due to the armature resistance; the product of the armature current and the voltage drop in the armature. The power loss is equal to $I_a^2 \times R$, and, therefore, this loss will increase as the load increases.

The **constant losses** are due, first, to friction and windage, and second, to hysteresis and eddy currents (called "iron losses"). These losses may be considered constant at all loads, provided that the speed is constant.

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When a motor is operated without load, the watts input equals the sum of the losses. If the resistance of the armature is known, the $I_o{}^2R$ losses can be computed, by simply measuring the armature current. It follows that

$$\text{Constant losses} = E(I_o + I_f) - I_o{}^2R.$$

In many cases the value of $I_o{}^2R$ will be so small as to be negligible.

The watts input, under load, is $E(I_a + I_f)$. Substituting the values of input and losses in the equation for brake horsepower,

$$746 \text{ B.hp.} = E(I_a + I_f) - I_a{}^2R - E(I_o + I_f) + I_o{}^2R$$

$$= EI_a + EI_f - I_a{}^2R - EI_o - EI_f + I_o{}^2R$$

$$= I_a(E - I_aR) - I_o(E - I_oR)$$

$$\text{B.hp.} = 0.00134[I_a(E - I_aR) - I_o(E - I_oR)].$$

(a) **Determination of the Horsepower Output.** In the above equation I_o and R are taken as constant and may be predetermined. Then, at any load applied to the motor, it is only necessary to measure the armature current, I_a, corresponding to that load, and the voltage, E, in order to calculate the brake horsepower.

The armature resistance, R, may be determined by the "drop of potential" method. With the field disconnected, connect the armature leads to the line with a *resistance* and ammeter in series, *being careful to avoid injury to the coils with excessive current*. Measure the current flowing, and the electromagnetic force drop across the brushes. Then, by Ohm's law,

$$R = \frac{\text{Voltage drop}}{\text{Amperes}}.$$

This should be repeated for a number of different positions of the armature, and the average resistance from the various positions taken. Repeat, also, for another value of current.

Next, the motor should be connected as in service, but with an ammeter in the armature circuit, and a voltmeter to indicate the electromagnetic force at the armature terminals. It is then run at no-load for a period of *30 min.*, in order to obtain uniform conditions of friction, etc. If, now, the ammeter is read, the required value of I_o is obtained.

If the motor is equipped with a rheostatic control, giving different speeds, the no-load value of I_o should be obtained for *each different speed* at which the motor is to be run, and a speed-ampere curve drawn. When

the motor is run under load, it is necessary to read not only the armature current, but also the speed in order to apply the corresponding value of I_o in the equation for brake horsepower.

(b) Efficiency. The total input to the motor is the product of the electromagnetic force and the sum of armature and field currents.

$$\text{Watts input} = E(I_a + I_f).$$

This value divided into the watts output equals the efficiency of the motor. It is sometimes convenient to plot efficiency against load at various speeds.

88. Test of a Direct Current Generator

Principles. Many prime movers are connected to generators and in order to determine that actual shaft horsepower developed by the machine, it is necessary first to determine the efficiency of the generator. Electrical engineers speak of two kinds of efficiency: *electrical efficiency* and *commercial efficiency*. **Electrical efficiency** takes into account only electrical losses while the **commercial efficiency** considers all losses and is therefore the efficiency to be used in connection with mechanical tests.

The output of the generator is the product of the terminal electromagnetic force and the current in the line, I_l. This current is not the same as the armature current, I_a, since part of the armature current goes to excite the field of the generator, $I_a = I_l + I_f$. Since generators are essentially constant speed machines, the losses may be treated in much the same manner as for a motor (see Test 87). The variable loss will be $I_a^2 R$, as before. The constant losses will be friction, windage, iron losses, and watts to field excitation, EI_f.

(a) Separation of the Constant Losses. The sum of windage and friction may be found by driving the generator with the field disconnected and the armature circuit open. In order to be sure that there is no small electromagnetic force, due to residual magnetism, a voltmeter should be connected to the terminals of the generator. If there is an indication of such residual electromagnetic force, it may be neutralized by giving the field a momentary excitation in the opposite direction from some outside source. The power required to drive the generator at its rated speed, under these conditions, is friction and windage. This, of course, presupposes a known source of power.

The iron losses cannot be separately determined, but the friction and windage having been found, they can be determined by difference. To

do this, the armature circuit is left open and the field is excited from an outside source. The normal field current is used in this test. The power required to drive the generator under these conditions is the sum of friction, windage and iron losses. Frequently only this determination is needed since it is not necessary to separate the losses in order to find the efficiency. These losses can be determined once and for all since the generator is a constant speed machine.

(b) Efficiency. The output of a generator is measurable and if the constant losses have been determined, the input to the generator is the sum of the output plus the variable loss plus the constant losses. The efficiency is of course the output divided by the input.

The subject of the tests of alternating current machines is too involved to be contained in a work of this scope. The reader should consult any standard work on alternating current machinery.

APPENDIX A

LOGARITHMS

To Find the Fractional Power of a Number Less Than Unity.
To avoid the use of negative characteristics, the following rule is suggested.
Rule. Express the given number as a fraction whose numerator is unity. Find the required power of the denominator of this fraction, and then reduce to a decimal.
For example,

$$\text{To find } 0.787^{1.33},$$

$$0.787 = \frac{1}{1.27}$$

$$\text{Log } 1.27^{1.33} = 1.33 \times 0.104 = 0.138;$$

$$0.138 = \log 1.37;$$

$$\therefore \quad 1.27^{1.33} = 1.37,$$

and

$$0.787^{1.33} = \frac{1}{1.27^{1.33}} = \frac{1}{1.37}$$

$$= 0.73.$$

To Find the Napierian Logarithm (Base e = 2.718) of a Number.
Rule. Multiply the common logarithm (base 10) by the constant 2.302, or 2.3, approximately.
For example,

$$\text{To find } \log_e \text{ of } 315,$$

$$\text{Log}_{10} \, 315 = 2.498,$$

$$\therefore \quad \text{Log}_e \, 315 = 2.3 \times 2.498 = 5.750.$$

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COMMON LOGARITHMS

No.	0	1	2	3	4	5	6	7	8	9	Dif.
0	0	0000	3010	4771	6021	6990	7782	8451	9031	9542	
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	42
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	38
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	35
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	32
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	30
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	28
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	26
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	25
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	24
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	22
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	21
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	20
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	19
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	19
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	18
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	17
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	16
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	16
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	15
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	15
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	14
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	14
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	13
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	13
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	13
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	12
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	12
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	12
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	11
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	11
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	11
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	10
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	10
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	10
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	10
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	10
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	9
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	9
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	9
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	9
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	9
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	9
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	8
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	8
54	7324	7332	7340	4348	7356	7364	7372	7380	7388	7396	8

COMMON LOGARITHMS

No.	0	1	2	3	4	5	6	7	8	9	Dif.
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	8
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	8
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	8
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	7
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	7
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	7
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	7
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	7
65	8129	8135	8142	8149	8156	8162	8169	8176	8182	8189	7
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	7
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	6
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	6
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8636	6
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	6
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	6
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	6
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	6
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	6
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	5
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	5
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	5
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	5
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	5
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	5
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	5
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	5
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	5
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	4

DIAMETERS AND AREAS OF CIRCLES

Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area
$\frac{1}{16}$	.00307	$\frac{15}{16}$	6.78	$\frac{13}{16}$	26.5	$\frac{3}{8}$	102.	$\frac{1}{8}$	230.
$\frac{1}{8}$	.0123	3.	7.07	$\frac{7}{8}$	27.1	$\frac{1}{2}$	104.	$\frac{1}{4}$	234.
$\frac{3}{16}$	.0276	$\frac{1}{16}$	7.37	$\frac{15}{16}$	27.7	$\frac{5}{8}$	106.	$\frac{3}{8}$	237.
$\frac{1}{4}$	.0491	$\frac{1}{8}$	7.67	6.	28.3	$\frac{3}{4}$	108.	$\frac{1}{2}$	240.
$\frac{5}{16}$	.0767	$\frac{3}{16}$	7.98	$\frac{1}{8}$	29.5	$\frac{7}{8}$	111.	$\frac{5}{8}$	244.
$\frac{3}{8}$	.110	$\frac{1}{4}$	8.30	$\frac{1}{4}$	30.7	12.	113.	$\frac{3}{4}$	247.
$\frac{7}{16}$	.150	$\frac{5}{16}$	8.62	$\frac{3}{8}$	31.9	$\frac{1}{8}$	115.	$\frac{7}{8}$	251.
$\frac{1}{2}$	.196	$\frac{3}{8}$	8.95	$\frac{1}{2}$	33.2	$\frac{1}{4}$	118.	18.	254.
$\frac{9}{16}$	.248	$\frac{7}{16}$	9.28	$\frac{5}{8}$	34.5	$\frac{3}{8}$	120.	$\frac{1}{8}$	258.
$\frac{5}{8}$	.307	$\frac{1}{2}$	9.62	$\frac{3}{4}$	35.8	$\frac{1}{2}$	123.	$\frac{1}{4}$	262.
$\frac{11}{16}$	.371	$\frac{9}{16}$	9.97	$\frac{7}{8}$	37.1	$\frac{5}{8}$	125.	$\frac{3}{8}$	265.
$\frac{3}{4}$	.442	$\frac{5}{8}$	10.3	7.	38.5	$\frac{3}{4}$	128.	$\frac{1}{2}$	269.
$\frac{13}{16}$	.518	$\frac{11}{16}$	10.7	$\frac{1}{8}$	39.9	$\frac{7}{8}$	130.	$\frac{5}{8}$	272.
$\frac{7}{8}$	.601	$\frac{3}{4}$	11.0	$\frac{1}{4}$	41.3	13.	133.	$\frac{3}{4}$	276.
$\frac{15}{16}$	.690	$\frac{13}{16}$	11.4	$\frac{3}{8}$	42.7	$\frac{1}{8}$	135.	$\frac{7}{8}$	280.
1.	.785	$\frac{7}{8}$	11.8	$\frac{1}{2}$	44.2	$\frac{1}{4}$	138.	19.	283.
$\frac{1}{16}$	.887	$\frac{15}{16}$	12.2	$\frac{5}{8}$	45.7	$\frac{3}{8}$	140.	$\frac{1}{8}$	287.
$\frac{1}{8}$	.994	4.	12.6	$\frac{3}{4}$	47.2	$\frac{1}{2}$	143.	$\frac{1}{4}$	291.
$\frac{3}{16}$	1.11	$\frac{1}{16}$	13.0	$\frac{7}{8}$	48.7	$\frac{5}{8}$	146.	$\frac{3}{8}$	295.
$\frac{1}{4}$	1.23	$\frac{1}{8}$	13.4	8.	50.3	$\frac{3}{4}$	148.	$\frac{1}{2}$	299.
$\frac{5}{16}$	1.35	$\frac{3}{16}$	13.8	$\frac{1}{8}$	51.8	$\frac{7}{8}$	151.	$\frac{5}{8}$	302.
$\frac{3}{8}$	1.48	$\frac{1}{4}$	14.2	$\frac{1}{4}$	53.5	14.	154.	$\frac{3}{4}$	306.
$\frac{7}{16}$	1.62	$\frac{5}{16}$	14.6	$\frac{3}{8}$	55.1	$\frac{1}{8}$	157.	$\frac{7}{8}$	310.
$\frac{1}{2}$	1.77	$\frac{3}{8}$	15.0	$\frac{1}{2}$	56.7	$\frac{1}{4}$	159.	20.	314.
$\frac{9}{16}$	1.92	$\frac{7}{16}$	15.5	$\frac{5}{8}$	58.4	$\frac{3}{8}$	162.	$\frac{1}{8}$	318.
$\frac{5}{8}$	2.07	$\frac{1}{2}$	15.9	$\frac{3}{4}$	60.1	$\frac{1}{2}$	165.	$\frac{1}{4}$	322.
$\frac{11}{16}$	2.24	$\frac{9}{16}$	16.3	$\frac{7}{8}$	61.9	$\frac{5}{8}$	168.	$\frac{3}{8}$	326.
$\frac{3}{4}$	2.40	$\frac{5}{8}$	16.8	9.	63.6	$\frac{3}{4}$	171.	$\frac{1}{2}$	330.
$\frac{13}{16}$	2.58	$\frac{11}{16}$	17.3	$\frac{1}{8}$	65.4	$\frac{7}{8}$	174.	$\frac{5}{8}$	334.
$\frac{7}{8}$	2.76	$\frac{3}{4}$	17.7	$\frac{1}{4}$	67.2	15.	177.	$\frac{3}{4}$	338.
$\frac{15}{16}$	2.95	$\frac{13}{16}$	18.2	$\frac{3}{8}$	69.0	$\frac{1}{8}$	180.	$\frac{7}{8}$	342.
2.	3.14	$\frac{7}{8}$	18.7	$\frac{1}{2}$	70.9	$\frac{1}{4}$	183.	21.	346.
$\frac{1}{16}$	3.34	$\frac{15}{16}$	19.1	$\frac{5}{8}$	72.8	$\frac{3}{8}$	186.	$\frac{1}{8}$	350.
$\frac{1}{8}$	3.55	5.	19.6	$\frac{3}{4}$	74.7	$\frac{1}{2}$	189.	$\frac{1}{4}$	355.
$\frac{3}{16}$	3.76	$\frac{1}{16}$	20.1	$\frac{7}{8}$	76.6	$\frac{5}{8}$	192.	$\frac{3}{8}$	359.
$\frac{1}{4}$	3.98	$\frac{1}{8}$	20.6	10.	78.5	$\frac{3}{4}$	195.	$\frac{1}{2}$	363.
$\frac{5}{16}$	4.20	$\frac{3}{16}$	21.1	$\frac{1}{8}$	80.5	$\frac{7}{8}$	198.	$\frac{5}{8}$	367.
$\frac{3}{8}$	4.43	$\frac{1}{4}$	21.6	$\frac{1}{4}$	82.5	16.	201.	$\frac{3}{4}$	371.
$\frac{7}{16}$	4.67	$\frac{5}{16}$	22.2	$\frac{3}{8}$	84.5	$\frac{1}{8}$	204.	$\frac{7}{8}$	376.
$\frac{1}{2}$	4.91	$\frac{3}{8}$	22.7	$\frac{1}{2}$	86.6	$\frac{1}{4}$	207.	22.	380.
$\frac{9}{16}$	5.16	$\frac{7}{16}$	23.2	$\frac{5}{8}$	88.7	$\frac{3}{8}$	211.	$\frac{1}{8}$	384.
$\frac{5}{8}$	5.41	$\frac{1}{2}$	23.8	$\frac{3}{4}$	90.8	$\frac{1}{2}$	214.	$\frac{1}{4}$	389.
$\frac{11}{16}$	5.67	$\frac{9}{16}$	24.3	$\frac{7}{8}$	92.9	$\frac{5}{8}$	217.	$\frac{3}{8}$	393.
$\frac{3}{4}$	5.94	$\frac{5}{8}$	24.8	11.	95.0	$\frac{3}{4}$	220.	$\frac{1}{2}$	398.
$\frac{13}{16}$	6.21	$\frac{11}{16}$	25.4	$\frac{1}{8}$	97.2	$\frac{7}{8}$	224.	$\frac{5}{8}$	402.
$\frac{7}{8}$	6.49	$\frac{3}{4}$	26.0	$\frac{1}{4}$	99.4	17.	227.	$\frac{3}{4}$	406.

DIAMETERS AND AREAS OF CIRCLES.—*Continued*

Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area	Diam.	Area
⅞	411.	⅜	466.	⅞	525.	⅜	588.	⅞	654.
23.	415.	½	471.	26.	531.	½	594.	29.	660.
⅛	420.	⅝	476.	⅛	536.	⅝	599.	⅛	666.
¼	425.	¾	481.	¼	541.	¾	605.	¼	672.
⅜	429.	⅞	485.	⅜	546.	⅞	610.	⅜	677.
½	434.	25.	491.	½	551.	28.	616.	½	683.
⅝	438.	⅛	495.	⅝	556.	⅛	621.	⅝	689.
¾	443.	¼	501.	¾	562.	¼	627.	¾	695.
⅞	448.	⅜	505.	⅞	567.	⅜	632.	⅞	700.
24.	452.	½	511.	27.	573.	½	638.	30.	707.
⅛	457.	⅝	515.	⅛	577.	⅝	643.		
¼	462.	¾	521.	¼	583.	¾	649.		

WEIGHT OF ONE CUBIC FOOT OF WATER AT VARIOUS TEMPERATURES

Temp., Deg. F.	Weight, Lb. per Cu. Ft.	Temp., Deg. F.	Weight, Lb. per Cu. Ft.
30	62.42	190	60.36
40	62.43	200	60.12
50	62.42	210	59.88
60	62.37	220	59.63
70	62.30	230	59.37
80	62.22	240	59.11
90	62.11	250	58.83
100	62.00	260	58.55
110	61.86	270	58.26
120	61.71	280	57.96
130	61.55	290	57.65
140	61.38	300	57.33
150	61.20	310	57.00
160	61.00	320	56.66
170	60.80	330	56.30
180	60.58	340	55.94

APPENDIX B

REPORTS OF ENGINEERING TESTS

A complete report should describe concisely the experimental object and how it was accomplished, and it should give numerical results and the conclusions formed from them. Whether the report is upon work performed by a student in the laboratory, or by an engineer in practice, its material may be arranged to advantage under the following heads:

1. Brief statement of object, results, conclusions, and recommendations.

2. Complete presentation of the leading facts regarding the entire work.

(a) Object, giving authority, preliminary agreements and other relevant information.

(b) Description and principles of apparatus tested, including sketches and photographs.

(c) Methods of testing, giving conditions of operation, location, arrangement and method of using instruments and testing apparatus, and personnel.

(d) Discussion of the data and results both as to accuracy and their bearing on the object of the test.

(e) Leading conclusions.

(f) Tables of data and calculated results or graphical expositions of the same.

(g) Charts not included under item (f).

3. Appendices giving details which are not included in (b) such as methods of calculation, methods of calibration, descriptions of special testing apparatus, results of preliminary or special tests, etc.

The subjects under these sub-heads may be treated as follows:

1. Object of the Experiment. This should be a clear, complete, and concise statement, preferably in one sentence. Its purpose, in practice, is to enable the reader to decide without a full reading whether or not the contents of the report come within his scope of interest. In student work, it should be included as a matter of training. In any case, the object of the test should be stated in writing before its performance in order that the experimenter and others concerned shall have a clear view of the undertaking.

In general, the statement of the object should be followed by a brief statement of the results obtained, the major conclusions reached and important recommendations. This serves much the same purpose as the statement of the object, in making a full reading of the report unnecessary in order to find whether or not it meets the needs of the reader.

2. Complete Presentation of the Leading Facts Regarding the Entire Work.

(a) Object. In reports of considerable magnitude, this portion states the object, authority for making the tests, preliminary agreements as to operating

conditions, etc., and any other relevant information. In reports of lesser magnitude, such as reports of student tests, this section can usually be omitted.

(b) **Description and Principles of Apparatus Tested.** This should deal only with the machine, instrument, material, or apparatus *tested*. The extent of the treatment is governed by the requirements of those for whom the report is intended. A report prepared as a technical article or address kills itself if it talks over the heads of its audience or bores them with details with which they are familiar, however much satisfaction it may give the author. This is a matter for judgment. A good rule to follow is to give complete descriptions only of new or little known apparatus; for others it is sufficient to give only commercial sizes and names. It should be remembered, however, that any distinctive feature or any characteristic affecting the test results should be fully described. This depends upon the object of the test. For example, in the report of a mechanical efficiency test of a steam engine, it is appropriate to describe thoroughly anything affecting friction in operation, such as lubrication details, balancing of valves, etc. But if the same engine were tested for its steam distribution, lubrication need not be mentioned at all; the valve mechanism then being the important item.

In this division there may be deduced any formulas used in getting results provided they are original or unusual. Otherwise, they may be quoted without deduction, with reference to the authority.

This section of the report should also contain sketches or photographs of the set-up of the apparatus. In general a diagrammatic sketch, which indicates the points at which readings were taken, is to be preferred. This, however, may be supplemented with photographs to good advantage.

(c) **Method of Testing.** A logical way to begin this subject is by reference to the formulas for the quantities sought. Each formula, reduced to the desired form, shows the quantities that must be directly measured. The means of measuring them may then be described. Usual instruments may be merely mentioned by name, but special apparatus (original, or applicable only to the particular test) should be described fully. Any limitations of apparatus or unusual facilities affecting the precision of measurements should be mentioned to enable the reader to judge for himself the accuracy of the results. For the same reason, the duration of the test should be stated with such items as frequency of readings. In this connection it is a good plan to refer to a sample set of observations which may be included in division (f).

This section of the report should also contain a list of the persons connected with the test and in many cases it is also advisable to state what each person did.

(d) **Discussion.** This portion of the report should deal with the probable accuracy of the results as affected by the precision of the instruments and methods used and as indicated by the concordance of the results; it should compare the results with corresponding ones from similar tests, records of which are available in handbooks or elsewhere.

(e) **Leading Conclusions.** Under this heading should be set forth the conclusions to be drawn relative to the performance of the apparatus tested and to the physical laws controlling the performance. The conclusions to be drawn from a test are the most important part of experimentation, in fact, its very *raison d'être*; so they should be fully given in the report. In practice, there

would be no value to a test from which conclusions could not be, or were not, drawn.

(f) **Tables of Data and Curves.** Numerical results should always be presented in the form of curves when possible, and also as a table. This enables the busy reader to size up the report without fully reading it. Curves and tables should have definite titles and enough information to explain their meaning without reference to the text. Curve sheets should bear the scales of coordinates, and the axes should be scaled for ready reading. Where several curves appear on one sheet, their plotted points should be differentiated by such conventions as circles in outline, solid, half solid, etc. The points should be clearly marked so that they will not be obliterated when the curve is drawn in.

The student should take care to differentiate between *results* and *observations*. Sometimes observations are results, but this is seldom the case. Further, there may be many intermediary quantities between the two which should not be confused with results. If the object of the experiment is clearly stated, it will always show what should comprise the results; in case of doubt, it should be referred to.

(g) This part of the report contains all auxiliary diagrams, such as graphical logs, flow sheets, etc. It would only appear in a report of considerable magnitude.

3. Appendices. Complete reports should contain tables giving the observations in full so that their concordance and validity may be checked by the scientific investigator. When this is not considered necessary, a sample set may be submitted, as for division 3. Students should include in reports the original notes taken in the laboratory, a loose-leaf note-book being used for convenience.

Rough notes should be taken neatly and contain enough data not only to remind the experimenter of the noted quantities, but to enable an outsider without additional explanation to interpret them fully. Inexperienced observers, to save time, are prone to use arbitrary symbols or abbreviations in their notes, without meaning to any one but themselves. The objections to this practice are that the notes cannot be checked by others, and the observer himself is apt to forget their meaning. Other items which may be included in this section of the report are sample calculations, method of calibration, descriptions of special testing apparatus, etc.

In practice, sample calculations should always be included to make clear the methods employed and as a voucher of the accuracy of the calculations. They should be brief, proceeding at once from the expression for the desired quantity, in which is substituted the test data, to the numerical result.

The student should bear in mind that *sample* means something *representative of the whole*. Therefore, if five different quantities are tested for, three times for each one in the same way, five samples are necessary and sufficient. If these tests were repeated by another method involving a different calculation, ten samples are needed. If only one determination is made of each quantity, the sample must be all of the calculations to be representative.

A METHOD FOR CONDUCTING STUDENT TESTS*

The following method has been used with success at Syracuse University.

The first laboratory work assigned is, as far as the equipment will allow, individual. After a little training in the methods of the work, the experiments are given as problems, the theory of which is taught in the classroom. In the laboratory the student is quizzed at intervals during his work, to insure that it be performed not as a matter of rote. Approximate calculations from observations are exacted as the observations are obtained. It is a great mistake to allow the postponement of such calculations until after the test is completed. This point cannot be too strongly emphasized. It is far better to have an experiment only partly but intelligently done than to have a vast amount of observations leading to faulty results and a half-grasp of the principles.

As the course advances, the tests, especially those upon large units, require more than one student to make all the necessary operations and measurements. For the successful conduct of some, it is expedient to have four or five stations at each of which one observer is needed. These advanced tests are undertaken in this way. In the classroom the students solve problems upon the principles involved and are given the special instructions which they could not reasonably be expected to obtain for themselves. When they have shown a satisfactory understanding of the principles, they are allowed to proceed with the experimental work, conducted as follows: For a test requiring five stations, for instance, a squad of six men is selected. For the last fifteen minutes or so of the test, observers A and B are at station No. 1, C at No. 2, D at No. 3, E at No. 4, and F at No. 5. Then B moves to station No. 2, where he is instructed by C in its duties. As soon as B has become familiar with them, C moves on to station No. 3 and is instructed by D concerning the work there, after which D moves to station No. 4 with E, and so on. The scheme can best be understood by the following schedule.

STATION NO.

Time	1	2	3	4	5
10:00–:15	AB	C	D	E	F
:15–:20	A	BC	D	E	F
:20–:25	A	B	CD	E	F
:25–:30	A	B	C	DE	F
:30–:35	A	B	C	D	EF
:35–:40	FA	B	C	D	E
:40–:45	F	AB	C	D	E
	etc.				

The advantages of this method over shifting all the students at one time are obvious. There is absolutely no break in the accuracy of the observations or continuity of the test, since only one man shifts at a time and he does not make measurements to be used before he has become somewhat familiar with the apparatus. The instructor is relieved of the small but essential details of instruction by the students themselves. Room is made for an additional man on the test without sacrificing the work of any of the others. A complete shift

* Abstracted from a paper by the present writer on "The Teaching of Experimental Engineering," *Educational Review*, June, 1912.

of every man can be made in a shorter time; in the example cited, the interval is 30 min.

The time necessary for an observer at a station to instruct the newcomer varies, of course, with the duties of the station. In a boiler test the average time is about 10 min., so that a complete shift of six men would be effected in 60 min. If all the men shifted at once, the interval should not be less than 90 min., preferably 2 hr.

The shifting is automatic. All that each student needs to know is the sequence of stations, and to remember that he is to move to the next one only when he has been relieved by, and has instructed his successor.

It has been found of considerable advantage to include a station at which a "General Log" is maintained. Observations are recorded from all the stations. The student in charge of it checks to some extent these observations upon recording them, and is required also to calculate roughly indicative results as the test proceeds. The log may be used for reference by all the students and enables a clearer view of the whole test.

At the end of a complete shift, three of the men are replaced by three new ones, and put upon the final calculations under the instructor's supervision. When these are completed the resulting quantities are plotted on a large chart, in common use for all who have made the test. In the meantime the test is continued by the three new and the three old observers, the arbitrarily varied quantity of the test having been changed. At the end of the next complete shift, half the men are again replaced, the ones remaining being the more recent ones. In this way each group of three men serves two complete shifts and makes the calculations from the observations of one. When all the results are in, their concordance is checked by the regularity of the plotted curves, and these are presented as a whole to the class for consideration.

APPENDIX C

PROBLEMS

Weights and Forces

1. Five weights of about 10 lb. each are to be standardized in order to calibrate a beam up to 50 lb. How closely should these secondary standards be weighed, if the least count of the beam to be calibrated is 4 oz.? (Note. Have regard for the cumulative error.)

2. Referring to division (c), how closely should the shot be weighed for less than 1 per cent of error in the determination of the leverage ratio?

3. The least count of a scales to be calibrated is 4 oz. Its leverage ratio is 100 : 1. How closely should the poise weights be weighed so that their calibration shall be within the precision of the beam? See (d).

4. Prove that it makes no difference in the instrument indications where W is placed on the platform.

5. If the rider, R, is too light, will the resulting error be constant at all indications of the beam or will it vary and why? Will the error be plus or minus?

6. Calculate the descending and combined scales for the example given by methods (a) and (b). Compare them.

7. Prove from the fact that the area under each curve equals work done that the descending curve cannot be a straight line.

8. From the ascending data of the example given, figure and compare the different results for the ascending spring scale obtained by the following (faulty) methods. $50 \div 1.68$. The average of all the F's $\div$ average of E's. The average of $F \div E$. The average of each increment of $F \div$ corresponding increment of extension. The average increment of F divided by average increment of E.

9. If the table and plunger of the test apparatus weigh 15½ oz., how much pressure does their weight produce in pounds per square inch, the diameter of the plunger being 0.50 in.? Is the difference between this and 5 psi. worth considering?

10. If the area of the plunger of the test apparatus is about ⅕ sq. in., how closely should the test weights be weighed to come within the precision of a gage having a least count of 5 lb.?

11. If a draft gage using water at 70° F. is correct, calculate the rise in temperature that would produce 1 per cent of error. *Ans.*, 60°.

12. In the Kent type of draft gage the reduced pressure inside the inverted can causes it to descend against the resistance of the spring (neglecting the buoyancy of the water). If the area of the inverted can is 50 sq. in. and if the pressure within is ¹⁄₁₀ in. of water less than without, what is the total downward force on the can? How much will the spring extend because of this force if the spring scale is 0.2 lb. per in.? *Ans.*, 0.9 in.

13. Deduce an equation for the gage, in Problem 12, neglecting buoyance, to show the relation between the extension of the spring and the draft, in inches of water, causing it.

Angular Velocity

14. Using an ordinary watch, how long should a continuous counter be timed when calibrating a tachometer with a range from 600 to 1200 r.p.m., at the even hundreds, if the least count is 10 r.p.m.? How long, if a stop watch is used?

Mechanical Power

15. What is the brake constant if $R = 5$ ft. $3\frac{1}{4}$ in.? If unbalanced weight of the brake is 2.5 lb., what scale reading would be necessary to balance 40 hp. at 160 r.p.m.?

Ans., 0.001; 252.5 lb.

16. By the third method the scales indicate 10 lb. when the brake is pulled up. (Fig. 46.) Its weight is not enough for it to drop, consequently the balance is reversed and the brake *pulled* down, for the weight − X reading. The scales then indicate 2 lb. What is the unbalanced weight? *Ans.,* 4 lb.

17. If the unbalanced weight is 5 lb., where should weight be added to balance the brake, and how much?

18. Given an arrangement like Fig. 46 except that the wheel turns anticlockwise and the spring balance is inverted. If the r.p.m. = 100, arm = 4 ft., unbalanced weight = 14 lb., and the balance reads 20 lb., what is the horsepower? *Ans.,* 2.59 hp.

19. The reading of a spring dynamometer running at 450 r.p.m. is 26°. If the spring constant is 3 lb.-in. per degree, and if the correction for windage, friction, etc., is 5°, how many foot-pounds of work will be done in 90 sec.? What will be the horsepower transmitted? *Ans.,* 22,200 ft.-lb.; 0.45 hp.

20. Same data as Problem 19, except that the friction of the transmission shaft is separately determined, the correction being 4°, and the correction for friction of the indicating device is 1°. What is the transmitted horsepower when the torque is increasing? When it is decreasing?

21. The axis of the piston p (Fig. 57) is 15 in. from the center of the driver shaft, and the diameter of the piston is 2 in. Disregarding friction, what fluid pressure is produced when the horsepower is 5, at 150 r.p.m.? *Ans.,* 44.0 psi.

22. Deduce the torque equation for Fig. 54.

Steam Engine Indicators

23. The pencil of an indicator throws 2 in. If the boiler pressure is 145 lb., what should the spring scale be to give as high a diagram as possible? *Ans.,* 80.

24. The ratio of the pencil motion to the piston motion of an indicator is 6 : 1. The area of the piston is $\frac{1}{2}$ sq. in. With a 30-lb. spring, the difference between ascending and descending positions of the pencil at a given pressure is 0.04 in. If the friction is all at the piston, how much is it in pounds? *Ans.,* 0.3 lb.

25. With the same indicator and spring at last problem, if the friction between the pencil point and the paper is 2 oz., how much difference in the height of the pencil will this make? *Ans.,* 0.05 in.

26. With the apparatus of Fig. 61, what is the effect of friction at the plunger upon the apparent true pressure? What effect has this upon the calibration records?

27. The force of acceleration of the indicator drum varies directly with its stroke and the square of the engine r.p.m. If a drum motion with a 4-in. stroke is designed correctly to operate at 200 r.p.m., how long should the stroke be to operate correctly at 250 r.p.m.? At 300 r.p.m.?   *Ans.,* 2.56 in.; 1.78 in.

28. What should be the constant of a drum spring (pounds per inch of extension measured on the card) to balance throughout a 4-in. stroke a force of acceleration which has a maximum value of + 1 lb. at the head end, and −1 lb. at the crank end?
                    Ans., 0.5 lb.

29. The stroke of the drum given by the reducing motion of Fig. 69E is 4 in. The ratio of the engine connecting rod to crank length is 5 : 1. Figure from the kinematic formulas the maximum error in the drum motion.

30. Is the best arrangement of Fig. 70C with the lower link always above the horizontal, always below it, or partly above and partly below? Why? What effect has the length of this link upon the accuracy of the motion?

31. Draw two indicator diagrams, superimposed, to show the effect of twisting the eccentric of Fig. 70D ahead of the crank by 10°.

32. Deduce the equation for the zero circle of a planimeter having the record wheel between the pivot and the tracing point instead of as shown by Fig. 38.

33. If a is 5 in.; b, 2 in.; and c, 3 in., what is the area of the zero circle applicable to an arm adjustment for a ¼ in. = 1 ft. scale?   *Ans.,* 2310 sq. ft.

34. The fixed center of a planimeter is inside an area to be integrated which is smaller than the zero circle; consequently the wheel rotates backward. The first reading is 29.23 sq. in.; the second, 48.73 sq. in. What is the area if the zero circle area is 212 sq. in.?   *Ans.,* 131.5 sq. in.

35. Integrate the area enclosed by a loop like a figure 8, first by finding separately the areas of the loop, and second by tracing the loop in the direction of its curve. Account for differences.

36. The arm c of a planimeter is 2.0 in. long and the diameter of its record wheel is 0.79 in. How many turns will the wheel make when the tracing point circumscribes 117 sq. ft. to a ¾-in. = 1 ft. scale?   *Ans.,* 13.1.

37. If the wheel diameter is 0.79 in., what should be the length of the arm c so that one revolution corresponds to 1000 sq. mi. on a linear scale of ⅛ in. = 1 mile?
                    Ans., 6.25 in.

38. Find the mean height of a given indicator diagram with a planimeter. Calculate the horsepower. Find the horsepower by adjusting the arm as described under (d). Compare results.

39. If the length of the arm c is 5 in., what should be the diameter of the wheel so that one revolution corresponds to 10 sq. in. of area?   *Ans.,* 0.45 in.

40. If the test under (a) gives results uniformly 8 per cent too small, is the arm too long or too short, and how much should it be changed if its length is 6.12 in.?
                    Ans., 0.45 in.

41. Using the indicator diagram of Problem 38, find the mean effective pressures by the Coffin planimeter, and compare results with those from the Amsler.

Flow of Fluids

42. Discuss the following observations taken from the test of a meter, and from them figure the true rate, the rate by the meter, and the correction factor.

Time	True Weight	Weight by Meter
2:10	50.5 lb.	10.0 lb.
2:15	68.8	20.8
2:20	80.1	31.5
2:25	91.3	42.1

The first true weight is that of an empty tank on a platform scales.

Ans., 2.25 and 2.13 lb. per min.; 1.06.

43. If a water meter of the volume or displacement type is accurate at all rates when the water is 60° F., draw its calibration and correction factor curves to be used when the water is at 120° F.

44. In calibrating a gas meter the capacity of which is 500 cu. ft. per hr., what rates should be applied, and how long should each trial be so that the error due to reading the gage glass is no more than 1 per cent? The diameter of the gasometer is 3 ft.

45. What should be the capacity of a gasometer to calibrate a meter of 2000 cu. ft. per hr. capacity?

46. Find the coefficient of discharge from the following experimental data obtained from a rectangular weir. Size of flume 18 ft. long by 3 ft. wide. Rate of rise, 0.12 ft. in 5 sec. Head on the weir, 5 in. Breadth of weir, 18 in. *Ans.*, 0.6.

47. What is the discharge in cubic feet per second from a nozzle having a diameter ratio of 2 to 1, the smaller diameter being ½ in., if the manometer shows 5 in. of mercury with its lower level 7 in. below the nozzle mouth? Assume $C = 0.97$.

Ans., 0.0246 cu. ft. per sec.

48. Assuming that a nozzle has been calibrated with water at 60°, will more or less water (weight and volume) emerge for a given reading of the U-tube when the water is at 120°? Why?

49. Given the diameter ratio 3 : 1; smaller diameter, 1 in.; find the constants K_1 and K_2 in the following equations:

$$Q = K_1 C \sqrt{H_1}$$

$$\text{Lb. per sec.} = K_2 C \sqrt{H_1}$$

in which H_1 is the difference of level in the mercury manometer in inches.

50. Draw the calibration for the meter of Problem 49, assuming that $C = 1$, up to a value of $H^1 = 6$ in.

51. Deduce the values given in the table under (a) for the distances of the stations from the pipe wall.

52. Give a rough value of the quantity rate in cubic feet of air per minute in a circular duct, 16 in. in diameter, in which the Pitot-static reading is 1.06 in. of water at the center of the duct. Air temperature is 70° F. *Ans.*, 4400.

53. Figure the constant for use in the quantity formula for an 8-in. pipe (internal diameter = 7.98 in.), the gaging liquid being mercury.

Ans., 0.285, in cu. ft. per sec. for 10 stations.

54. Figure the cubic feet per second in an 8-in. pipe (internal diameter = 7.98 in.) if a traverse gives the readings, 1.02, 1.26, 1.42, 1.56, 1.69, 1.65, 1.5, 1.38, 1.25, 1.01 in. of mercury. *Ans.*, 3.32 cu. ft. per sec.

55. What is the velocity, in cubic feet per second, and pounds per second of flow of illuminating gas in a 3-in. pipe (internal diameter = 3.07 in.) if the reading of the differential gage is 1.5 in. of oil of specific gravity 0.85? Pressure of the gas is 4 in. of water; barometer, 29.7 in. mercury; temperature 55° F. R, for this gas, is 60.7.

Ans., 0.276 lb. per sec.

56. Figure the constant in a formula like $V = 66.7 \sqrt{h}$, to apply to Problem 55. Take average conditions to be 14.7 lb. absolute pressure and 60° F. *Ans.*, 65.8.

57. What is the velocity in feet per second if $h = 12$ in. of water, pressure = 4 in. of mercury, temperature = 120° F., barometer = 30.1 in. of mercury, the gas being air. Figure answer by both approximate and accurate formulas.

Ans., 231 and 226 ft. per sec.

58. What is the coefficient of discharge if an orifice discharges 92 lb. per min. of water under a pressure of 3 lb. per sq. in.? Diameter of the orifice is 0.6 in.

59. If the diameter of the contracted section (see Problem 51) is 0.36 in., what are the coefficients of contraction and velocity? *Ans.*, 0.6; 0.99.

60. Write the equation of energy of air on the two sides of an orifice, neglecting kinetic energy at the entrance. Allowing a drop of pressure equal to 6 in. of water, compare the velocity due to pressure energy with that due to intrinsic energy assuming adiabatic flow.

61. Design an orifice and containing conduit to measure the air supplied to a 12-hp. gas engine taking about 20 cu. ft. of fuel gas per horsepower hour and about 12 cu. ft. of air to one of fuel. Drop of pressure through the orifice should not exceed 6 in. of water.

62. A blower discharges 2000 cu. ft. per min. of air through an 8-in. pipe. Design an exit conduit to reduce the velocity a proper amount so that the air may be measured with an anemometer, and show where the anemometer should be placed.

63. If the encircling frame of an anemometer is the same diameter as the opening through which gas is discharged, what should be the area with which to calculate quantity? Examine an anemometer to answer this question.

64. What should be the least count of an ammeter to measure 600 cu. ft. per min. of air under room conditions within 3 per cent of error? The voltage of the line is about 110. What should be the least count of the thermometers? Temperature rise of air to be about 20° F. *Ans.*, about 2 amp.

65. Using Napier's formula, what is the flow of steam in pounds per second through a ¼-in. diameter orifice, if the gage pressure is 30 lb.? What is the lowest pressure to which this method applies if the steam discharges into the atmosphere?

Ans., 0.0314 lb. per sec.; 25-lb. gage.

66. Using a chart record from an actual steam meter, figure the average rate by taking off velocities at a number of equal time intervals. Figure the average rate from the mean height obtained by a planimeter. Compare results.

Thermometry

67. A thermometer reads 240° when it is immersed to the 60° graduation. If temperature of room is 80°, what is the actual measured temperature? Estimate the stem temperature as an average of room and indicated temperatures. *Ans.*, 241.4°.

68. What should be the least count of a pressure gage to calibrate by steam a thermometer whose least count is 2°? Why? *Ans.*, 1 lb. at 300°.

69. Account for the differences between up and down readings in the calibration of a thermoelectric couple.

Calorimetry

70. Pressure of steam in the calorimeter chamber is 1.5 in. of mercury. Barometer is 30.3 in. of mercury. If the steam were saturated in the chamber, what would the temperature be? If the thermometer indicates 280°, how many degrees of superheat are there? What is the total heat of the steam in the calorimeter at 280°? If the pressure of the incoming steam is 84 lb. gage, what is its quality?   *Ans.*, $x = 0.995$.

71. Figure the quality from the data given in Problem 70, assuming the calorimeter pressure to be equal to atmospheric, 14.7 lb. What is the percentage of error?

72. Figure the quality of steam corresponding to superheats of 25, 45, and 65°, using the approximate method of constants for Problem 70.   *Ans.*, 0.972, etc.

73. What is the maximum degree of superheat that it is possible for the steam in the calorimeter to attain if the steam pressure is 95-lb. gage, and the calorimeter pressure 14.7 lb. abs.? What is the maximum if the steam pressure is 65-lb. gage?

74. What is the maximum percentage of moisture that can be shown by a calorimeter under the two conditions named in Problem 73? If the calorimeter pressure is reduced to 10 lb. abs. by connection with a condenser, what then is the maximum percentage of moisture?

75. If a thermometer reads 3° low on account of radiation, what is the percentage of error resulting from ignoring the radiation correction? Use the data of Problem 70.
                                                                                                             Ans., 0.17%.

76. If the water gage has been calibrated using the water separated from steam at 100-lb. gage, what percentage of error will there be when the instrument is used for steam at 50-lb. gage, the error being due to the change in density of the water between the two temperatures? If the water in the gage glass is 150° F., due to radiation, and the water in the calorimeter chamber is 325° F., will the level in the glass be above or below the inside level? Will these errors make material error in the results of x and why?   *Ans.*, 2.4%.

77. If the orifice of the separating calorimeter is used as part of a throttling calorimeter so as to allow for moisture escaping the separator, deduce a formula for the quality of the steam in terms of the weights shown by the separating calorimeter and the quality (x_1) as shown by the throttling calorimeter.

78. The data from a barrel calorimeter determination are as follows: Weight of water in barrel before condensing = 350 lb., temperature = 550° F. Weight after condensing = 370.6 lb., temperature = 115°. Pressure of steam sample = 100-lb. gage. What is the quality, neglecting the water equivalent and radiation?   *Ans.*, 0.991.

79. A water equivalent determination for the preceding example shows that 300 lb. of water will fall from 98.6° to 97.3° F. when placed in the barrel at 60°. What is the water equivalent, and what is the corrected value of the quality? *Ans.*, 10 lb.; 0.934.

80. The circulating water in a surface condenser rises in temperature from 50° to 86° F., 275 lb. being used to condense 10 lb. of a steam sample at 80 lb. gage. The temperature of the condensate is 98°. What is the quality of the sample?
                                                                                                             Ans., 0.856.

Constituents of Fuels

81. The proximate analysis of a coal gives moisture = 1 per cent, volatile matter = 22, fixed carbon = 72, and ash = 5. What are the percentages of hydrogen and

carbon in the volatile matter, and what is the percentage of total carbon? Base answers on coal as analyzed, not on combustible. *Ans.,* 4.6, 8.5, 80.5%.

82. Give the proximate analysis in the preceding problem on the basis of "dry" coal instead of coal "as received." *Ans.,* 1.01%, etc.

83. A sample of bituminous coal is tested for heat value. Ignition by hot wire. 2.5 liters of water used. Water equivalent of calorimeter = 0.40 lb. Sample weighs 0.6 g. How much heat is generated and what is the heat value if temperature rise is 4.23° F.? *Ans.,* 13,300 B.t.u.

84. Calculate the heat value of coal giving a proximate analysis as follows: moisture, 2.75; volatile matter, 6.00; fixed carbon, 78.45; ash, 12.8 per cent. Use Mahler's curve. What would be the heat value if the coal were dry?

Ans., 12,900 and 13,300 B.t.u.

85. Figure the higher heat value from a calorimeter test giving data as follows: Pressure of gas = 4 in. of water; barometer, 29.4 in.; temperature of gas, 70° F.; 7.24 lb. of water raised from 60.7° to 116.3°. Volume of gas burned, 0.875 cu. ft. by meter. *Ans.,* 500 B.t.u.

86. In the preceding problem, if there were 0.48 lb. of water of condensation, what would be the lower heat value? *Ans.,* 437 B.t.u.

Steam Engines

87. Following are the data from a test for volumetric clearance by the water method. Weight of bucket and water before filling 12 lb. 14¾ oz. Weight after filling 10 lb. 2½ oz. Time of filling, 95 sec. To keep clearance space full for 60 sec., 8 oz. What is the average leakage rate in pounds per minute? What is the clearance in cubic inches allowing for leakage? What is the per cent of clearance if the engine is 10 in. × 16 in. (bore × stroke)? *Ans.,* 5.22%.

88. In the preceding problem room temperature 60° F. of the water is assumed. How much percentage of error would be involved if the actual temperature were 40° F.? If 80°? *Ans.,* 0.1%, 0.3%.

89. The valve in the preceding has an eccentricity of 1.375 in. Find the lead and the maximum port opening on the crank end for equal cutoffs, using the Bilgram diagram. The connecting rod is 28 in. long and the crank 5 in.

90. Find the events of the stroke in per cents, using the data of the last problem and the Bilgram diagram.

91. Repeat the last problem, using the Zeuner diagram.

92. Why cannot a valve be set for equal leads and equal cutoffs at the same time? If for equal leads, which cutoff is greater?

93. How could the events of the stroke be measured direct from an engine whose valve was set, by reference to the valve and measurements of the crosshead travel?

94. Set the valve of an engine for equal cutoffs. Then measure the events of the stroke from the engine, from the indicator diagram actually obtained, and from the Bilgram diagram.

95. The lead on the head end of a steam engine is found to be $1\frac{1}{16}$ in., and on the crank end $\frac{3}{16}$ in. If the leads are to be made equal to $\frac{1}{16}$ in., should the valve stem be lengthened or shortened and how much? Should the angular advance be increased or decreased?

96. A D-slide valve has a face 1.5 in. long at each end, and the distance between the faces is 2.75 in. The ports are each $^{11}\!/_{16}$ in., and the outside edges of the ports are 4.50 in. apart. If the head end outside lap is 0.625 in., what are the values of the other three laps? *Ans.*, $\frac{5}{8}$, $\frac{3}{16}$, $\frac{3}{10}$ in.

97. Will increasing the length of a steam link increase or decrease the lap and the lead? Why? What effect will an increase of the angular advance have on laps and leads? Why?

98. If an indicator diagram shows the compression on the head end to be too early, should the exhaust link length be increased or decreased, and what effect will the readjustment have on the release?

99. An indicator diagram shows the cutoff to be one-fifth of the stroke. The pressure at cutoff is 110 lb. abs.; at the end of compression it is 25 lb. abs. The engine is 10 in. $\times$ 12 in. $\times$ 100 r.p.m., single acting with 4 per cent clearance. How many pounds of steam are supplied per hour, allowing 25 per cent of that shown by the diagram for condensation? *Ans.*, 233.

100. If the mean-effective pressure in the preceding is 25 lb., what is the steam consumption in pounds per I.hp.-hr.? *Ans.*, 39.2.

101. If the steam in the supply pipe (last problem) is at 105-lb. gage and contains 3 per cent of moisture, and if the exhaust is at 3 lb. gage, what is the thermal efficiency? *Ans.*, 6.69%.

102. What is the Rankine efficiency for the foregoing? *Ans.*, 48.1%.

103. Deduce from the typical form of Willan's line, the form of the curve between steam consumption in pounds per B.hp.-hr. and B.hp. What is the value of the steam consumption when B.hp. = 0?

104. If the F.hp. is constant at all values of the B.hp., deduce the forms of the curves of B.hp. vs. I.hp. and efficiency vs. B.hp. Will these curves pass through the origin or not? Why?

105. Figure what may be the maximum ratio of mean effective pressures with a 14-in. $\times$ 30-in. engine having a $2\frac{7}{16}$-in. piston rod, so that the error from using the approximate formula for I.hp. will be less than 1 per cent. *Ans.*, 5, nearly.

106. Calculate the length of the adjustable arm of a polar planimeter, so that horsepower may be read direct when indicating the engine in the preceding problem. The diameter of the record wheel is 0.79 in. and there are 100 graduations on the wheel. One graduation to equal 1 hp. Length of indicator diagram = 5 in. Spring scale = 60. Average value of N is 80. *Ans.*, 3.62 in.

107. A 50-horsepower engine running at 200 r.p.m. is tested with a brake with an 8-ft. arm. If its unbalanced weight is 10 lb. and the weight of the pedestal by which its thrust is transmitted to a platform scales is 5 lb., what should be the scale readings for an efficiency test of six runs? *Ans.*, 179 lb., max.

108. Run a series of tests on a steam engine to show the relation between steam pressure at admission if the governor is throttling, or cutoffs, if of the cutoff type, under variable brake horsepower.

Steam Turbine

109. A turbine, tested at full load, consumes 3150 lb. of steam in 90 min. The current delivered is 340 amperes and 220 volts. The generator efficiency at full load is 95 per cent and the friction losses of the turbine are 5 per cent of the power de-

livered to the generator. What is the steam consumption in pounds per kilowatt-hour, per E.hp.-hr., per B.hp.-hr., and per internal hp.-hr.?

Ans., 28.1 per Kw.-hr.

110. For the test of Problem 190, average values as follows were obtained. Steam pressure, 150-lb. gage; superheat, 80° F.; vacuum, 20 in. mercury; barometer, 28.5 in. What would have been the steam consumption if the following conditions held? Steam pressure, 180 lb. abs.; superheat, 100° F.; vacuum, 28 in.; barometer, 30 in. Corrections are 0.05 lb. per kw.-hr. for each pound difference in the steam pressure, 0.02 lb. for each degree of superheat, and 1.0 lb. for each inch of vacuum.

Ans., 20.5 lb. per kw.-hr.

111. What is the thermal efficiency for the results of Problem 109? How much efficiency should be added or subtracted for each unit of steam pressure, back pressure, and superheat. (See Problem 110.)

Hydraulic Turbine

112. What is the pressure energy in foot-pounds per minute available to a Pelton wheel supplied with 10 cu. ft. per min. of water at 100 lb. pressure?

Ans., 144,000 ft.-lb.

113. In the preceding problem, if the pipe supplying the water is 2 in. inside diameter, what is the velocity energy? What is the total horsepower?

Ans., 565 ft.-lb. per min.; 4.38 hp.

Internal Combustion Engines

114. An inlet valve is found to open too early and close too late. The exhaust valve on the same cylinder opens and closes too early. What should be done to correct the timing, if the two cams are integral with the cam shaft?

115. Draw two indicator diagrams, each superposed on a normal diagram like Fig. 106, to show the effects of too early and too late valve events.

116. Draw a suction diagram to be expected with normal setting when the reducing motion is set 90° ahead of the crank. Draw the work diagram.

117. A 2-cylinder, 6-in. × 8-in. × 350 r.p.m. gas engine, governing on the hit-and-miss principle, misses one out of every five explosion strokes. If the net mean-effective pressure is 20 and 21 lb. in the two cylinders, what is the I.hp.?

Ans., 3.28 hp.

118. What is the engine constant of a 6-in. × 8-in. gas engine? If the B.hp. is 5, and the M.e.p. of the upper loop of the diagram is 70, and of the lower loop 5 lb., and the engine fires 163 times per min., what is the horsepower lost to mechanical friction? To fluid friction? *Ans.*, 0.000571, 1 hp., 0.077 hp.

119. Compare by actual test the speed regulation of an engine working on the hit-and-miss principle with a throttling engine. Determine coefficient of regulation at full and half loads.

Steam Boiler

120. A 50-hp. boiler delivers steam at 75-lb. gage, superheated 100°, from feed water at 50°. What is the F.e.? How many pounds of feed water will be needed per hour during a test? Assuming an efficiency of 70 per cent and anthracite coal with 15 per cent ash, about how many pounds of coal will be needed per hour?

Ans., 1.27; 1360 and 190 lb.

121. For a given natural draft, there is a corresponding flue temperature necessary to maintain it. If the excess air during the operation of a boiler varies, and all other conditions remain constant, deduce the relation between per cent CO_2 and efficiency. Note that efficiency $= 1 -$ a constant $-$ loss to dry exhaust gases, approximately.

Pumps, Injectors, Etc.

122. The following data are obtained from the test of a duplex pump. Discharge pressure, gage, 35 lb.; suction lift, 4.5 ft.; water discharged per minute, 110 lb.; length of stroke, 5 in.; number of strokes per minute, both cylinders, 250; size of water cylinders, 2 in. $\times$ 5 in. What is the W.hp., the percentage of slip, and the capacity of the pump?   *Ans.*, 0.283 hp.; 22.6%.

123. In the preceding problem, if the internal diameter of the discharge pipe is 1.38 in., what is the horsepower due to velocity?

124. If the pressure in the discharge pipe is 12 lb., and if the water supply is 6 ft. above the gage indicating the discharge pressure, what is the total head?

Ans., 21.6 ft.

125. Run a series of tests on a steam pump to show the relation between slip and discharge pressure, and slip and speed.

126. The steam used for the test cited in Problem 122 was 88 lb. per hr., and was at 85-lb. gage pressure, quality 97 per cent. The pump exhaust steam was at 10-lb. gage. What were the steam consumption in pounds per horsepower-hour, the heat consumption in B.t.u. per horsepower-hour, the thermal efficiency and the duty?

Ans., Eff. $= 0.86\%$.

127. A pump discharges 4750 lb. of water against a total head of 100 ft., in a certain time. During the same time it is supplied with 100 lb. of steam and from each pound it consumes 1000 B.t.u. What is the duty of the pump?   *Ans.*, 4,750,000.

128. A test of an injector yields the following data: Steam pressure, 100 lb. gage; quality, 97 per cent; temperature of feed, 60° F.; temperature of discharge, 147° F.; pressure of discharge, 105 lb.; gage in discharge pipe, 5 ft. above injector; level of feed water, 10 ft. *above* injector; cross-section of supply tank, 7.1 sq. ft.; rate of fall of water in supply tank while pumping, 4.12 in. per min. What is the ratio of water pumped to steam used, the pounds of steam per W.hp.-hr., the efficiency, and the duty? Base all results on the total amount of discharge.   *Ans.*, Eff. $= 0.38\%$.

129. What should be the capacity of an injector to supply a 100-hp. boiler, if it operates only two-thirds of the time?

130. How would you determine separately the work done by the various impellers of a stage centrifugal pump?

131. Does or does not the measurement of total head by gages, outlined under (c), include the friction head of suction and discharge pipes, and why? Does it include the friction head of the water in passing through the pump, and ought this head to be included? Why?

Condensers, Feed-Heaters, Etc.

132. The gage on a condenser shows 3.45 in., abs. The temperature of the steam space is 115°. What percentage of air is present (based on the mixture)?

133. A condenser operating at 26-in. vacuum (normal barometer) takes cooling water at 60° and discharges it at 116°. What is the mean temperature difference?

Ans., 29.7°.

134. If in the preceding, 200,000 lb. of cooling water are used per hour, and the area of the cooling surface is 1000 sq. ft., what is the value of C?   *Ans.,* 377 B.t.u.

135. Using data of Problem 133, how many pounds of cooling water are required per pound of condensate, assuming dry steam and allowing 12% for heating air, radiation, etc.?   *Ans.,* 20.5 lb.

136. A jet condenser operates under a vacuum of 26 in. and takes condensing water at 70° F. Assuming the steam dry, and containing no air, how much water is required per pound condensate under ideal conditions?   *Ans.,* 18.2 lb.

137. A jet condenser operates under a vacuum of 26 in. and takes condenser water at 70° F. The temperature of the wet air pump discharge is 10° below that of the steam. Allowing 10 per cent for cooling the air, etc., how much water is required per pound of condensate from dry steam? (Compare with 136.)   *Ans.,* 24.7 lb.

138. A condensing plant is arranged so that the exhaust from the auxiliaries preheats the feed water after it leaves the condenser, in a closed heater. Tell how the plant could be tested so as to show the loss due to drop in the heat of the liquid between the engine exhaust and the heater entrance, and the gain due to the feed-water heater.

Refrigeration

139. What is the tonnage (ice-melting capacity) of a plant that circulates 6000 lb. of brine (specific heat = 0.80) per hour at an average temperature fall of 15° F.?

Ans., 6 tons.

140. What is the specific heat of a calcium chloride brine having a specific gravity of 1.2, if its temperature is 32° F.? If its temperature is 0° F.?   *Ans.,* 0.717; 0.701.

141. Supposing 600 lb. of anhydrous ammonia are circulated per hour at a head pressure of 160-lb. gage, and a suction pressure of 20-lb. gage, the superheat being 25°; what should be the refrigerating effect?   *Ans.,* 29,600 B.t.u. per hr.

142. A refrigerating plant uses 14,220 lb. per hr. of water in the ammonia condenser, with average incoming and outgoing temperatures of 50° and 75°, respectively. The I.hp. of the compressor is 20. Approximately what is the tonnage (ice-melting capacity)?   *Ans.,* 25.4 tons.

143. What is the actual coefficient of performance for the data of Problem 142?

Ans., 6.

Air Compressors and Blowers

144. In the operation of a blower, the volume discharged varies very nearly as the rotative speed. The pressure is due to centrifugal force. From these facts, deduce how the power will vary with the speed.

145. In the test of a blower, if gasoline, specific gravity = 0.75, is used as a gaging fluid with a Pitot tube, what is the velocity head in feet of air, if $h = 4.5$? Pressure is 6 in. of the same fluid. Barometer is 30.3 in. mercury. Temperature is 70° F.

Ans., 229 ft.

146. In the preceding, how many foot-pounds of work will be done in 1 min. if pipe diameter = 6 in.?   *Ans.,* 58,100.

147. An air compressor delivers 8.3 cu. ft. per min. of air at 100 lb. pressure abs., and 220° F. What is its capacity in cubic feet of free air per minute, temperature of the room being 75°, and pressure, 29.5 in. of mercury? What is its capacity in cubic feet of compressed air corrected to room temperature?

Ans., 45.1 and 6.52 cu. ft.

148. The compressor of the preceding problem was two-stage, 14-in. and 9-in. bore by 12-in. stroke, and 100 working strokes per min. per cylinder. What is the volumetric efficiency from the same data? *Ans.*, 42.2%.

149. What is the net air horsepower from data of Problem 147? What is the horsepower available in the compressed, cooled air? *Ans.*, 5.5 hp.

Miscellaneous Problems

150. The capacity of a ram operating against 20 lb. pressure is 1042 gal. per 24 hr., and its efficiency (Rankine) is 35 per cent. The supply level is 10.5 ft. above the ram. How much water passes through the waste valve in gallons per 24 hr.?
Ans., 10,100 gal.

151. Using the data of the preceding problem, what is the D'Aubisson efficiency?
Ans., 41%.

152. Assuming that the friction losses of a hoist are constant throughout its working range, deduce the form of, and sketch roughly, curves between force applied and load lifted and between efficiency and load lifted. Will these curves pass through the origin or not, and why?

153. A differential hoist has 12 chain notches on the driving wheel and 11 on the smaller. What is its ideal mechanical advantage? *Ans.*, 24.

154. The efficiency of a hoist is 60 per cent and its ideal mechanical advantage is 40. What load will a force of 4 lb. lift? *Ans.*, 96 lb.

155. Prove that a hoist having an efficiency greater than 50 per cent needs a locking device.

156. The torque as shown by the friction brake is 100 lb.-ft. If the follower pulley is 2 ft. in diameter, what is the difference between the belt tensions? If the sum of the belt tensions is 200 lb. and the angle of contact is 190°, what is the coefficient of friction? *Ans.*, $f = 0.33$.

157. Describe how a test should be made to show the variation of the coefficient with slip, all other conditions being constant.

158. Describe how a test should be made to show the variation of the coefficient with the normal force of contact, slip and other conditions being constant.

159. At no load, a motor takes 10.5 amperes armature current at 220 volts. What is the value of AV_0? Is this what is referred to as the "constant" loss? What is the constant loss, in watts (see Problem 161)? *Ans.*, 2310 watts.

160. Same motor, given the armature resistance $= 0.038$ ohm. What is the brake horsepower, when the armature current is 150 amperes? *Ans.*, 40 B.hp.

161. What is the efficiency of the preceding, if the field current is 5.8 amperes?
Ans., 87%.

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